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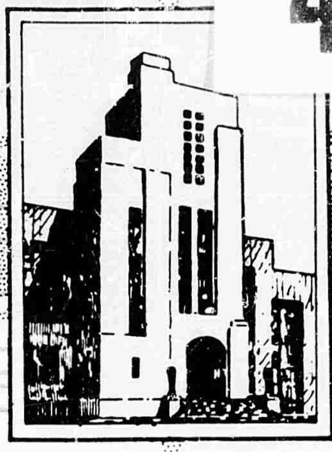


DEPARTMENT OF THE NAVY

DAVID TAYLOR MODEL BASIN

(5) David Taylor Model Basin, Washington, D.C

HYDROMECHANICS

AERODYNAMICS

STRUCTURAL MECHANICS

APPLIED MATHEMATICS

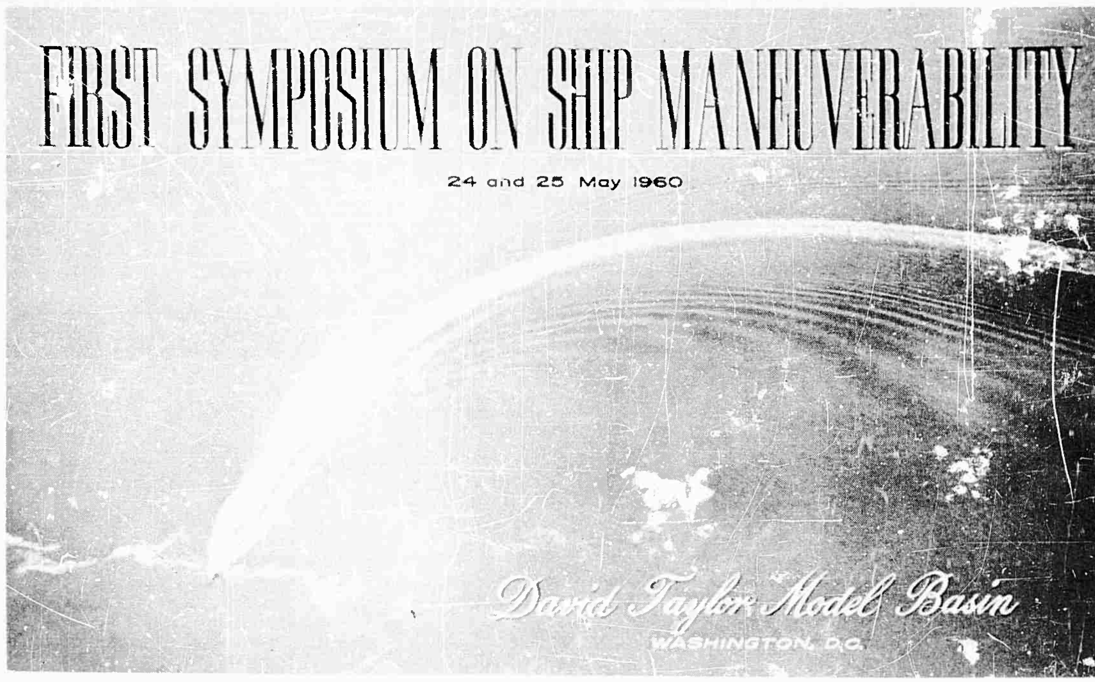
(6) FIRST SYMPOSIUM ON SHIP MANEUVERABILITY

DAVID TAYLOR MODEL BASIN

24 and 25 MAY 1960

October 1960

Report 1461



David Taylor Model Basin Report 1461

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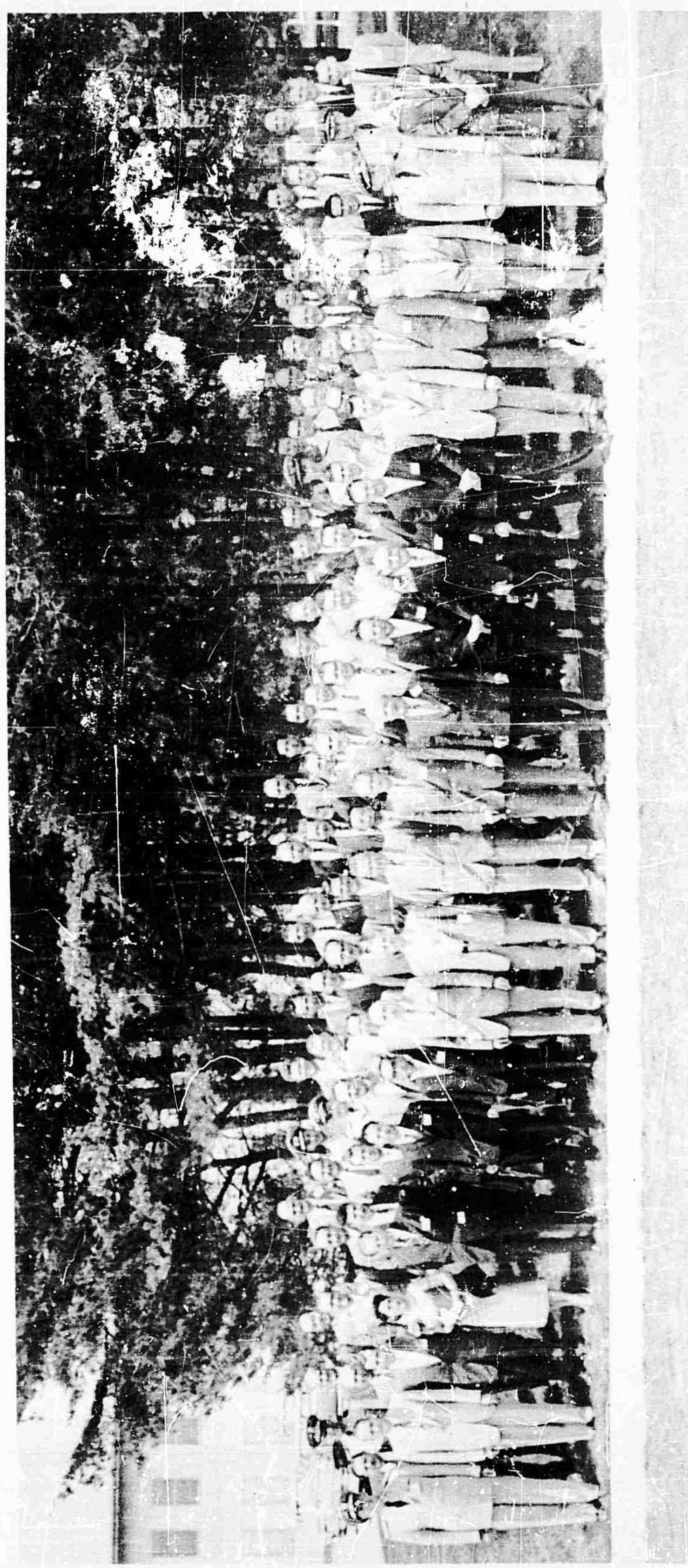
The First Symposium on Ship Maneuverability was motivated by the recent upsurge in importance of the subject to both the United States Navy and commercial shipping interests. In addition, there has been a rapid growth in laboratories having capabilities for doing research and development work in this area. It is hoped that the Symposium will serve to coordinate and supplement the fund of scientific information in a field of endeavor in which there are so few published works other than classified or proprietary reports.



First Symposium on Ship Maneuverability

ATTENDANCE LIST

Dr. Martin A. Abkowitz	Massachusetts Institute of Technology
Mr. A.Q. Aquino	David Taylor Model Basin
LT W.J. Aston	United States Naval Academy
Mr. A.L. Beal	David Taylor Model Basin
Mr. Bengt Bengtsson	Swedish Shipbuilding Experimental Tank
Mr. D.W. Bennett	Norfolk Naval Shipyard
Mr. Serge Bindel	Bassin d'Essais des Carenes, Paris, France
Mr. Berend J. Bracht	United States Coast Guard
Mr. F.D. Bradley	David Taylor Model Basin
Dr. John P. Breslin	Davidson Laboratory
Mr. D.F. Brown	David Taylor Model Basin
Mr. W.F. Brownell	David Taylor Model Basin
Mr. Otto Bussemaker	Hoofd V.H. Bureau Scheepsbouw, The Hague, Netherlands
Mr. A.S. Bussey	Bureau of Ships
Mr. Robert C. Case	United States Coast Guard
Mr. F.S. Cauldwell	General Dynamics Corporation, Electric Boat Division
Mr. E.P. Clement	David Taylor Model Basin
Mr. S.F. Crump	David Taylor Model Basin
Dr. W.E. Cummins	David Taylor Model Basin
Mr. L.M. DeLand	Bureau of Ships
Mr. E. Scott Dillon	Maritime Administration
Mr. M. Mack Earle	Maryland Shipbuilding and Dry Dock Company
RADM J.M. Farrin	Bureau of Ships
Mr. Samuel Feldman	Special Projects Office
Mr. Vincent Fighera	New York Naval Shipyard
Mr. P. Finkelstein	Long Beach Naval Shipyard
LCDR W.J. Flynn	United States Naval Academy
CAPT R.E. Foster	Bureau of Ships
Mr. M. Gertler	David Taylor Model Basin
CDR W.C. Gibson	United States Naval Academy
Mr. T.C. Gillmer	United States Naval Academy
Mr. Jerome L. Goldman	Friede and Goldman, Inc.
Mr. P. Golovato	David Taylor Model Basin
Mr. A. Goodman	David Taylor Model Basin
Mr. S.C. Gover	David Taylor Model Basin
Dr. W.L. Haberman	David Taylor Model Basin
Mr. G.R. Hagen	David Taylor Model Basin
Mr. Norman A. Hamlin	Bethlehem Steel Company
Mr. C.H. Hancock	Newport News Shipbuilding and Dry Dock Company
Mr. B.H. Hansen	Bureau of Ships
CAPT N.E. Harkleroad	David Taylor Model Basin
Mr. C.R. Herter, Jr.	Dravo Corporation
Mr. J.G. Hill	Bureau of Ships
Mr. Ludwig C. Hoffman	Maritime Administration
CDR P.A. Horn	David Taylor Model Basin
Mr. F.H. Imlay	David Taylor Model Basin
RADM R.K. James	Bureau of Ships
Mr. D.A. Jewell	David Taylor Model Basin
Mr. E.E. Johnson	David Taylor Model Basin
LCDR J.L. Jones	David Taylor Model Basin
Dr. P. Kaplan	Technical Research Group, Inc.
Mr. Raymond Kaufman	M. Rosenblatt and Son, Inc.
Dr. A.H. Keil	David Taylor Model Basin
Dr. Louis Landweber	Iowa Institute of Hydraulic Research State University of Iowa
Mr. S.W. Lank	United States Coast Guard
Mr. S.E. Lee	David Taylor Model Basin
Mr. E.V. Lewis	Davidson Laboratory
Mrs. May J. Lum	David Taylor Model Basin
Mr. Samuel M.Y. Lum	Bureau of Ships
Mr. S.T. Mathews	National Research Council, Canada
Mr. Robert Mende	Bird-Johnson, Company
Mr. W.F. Milliken	Cornell Aeronautical Laboratory
Mr. C.G. Moody	David Taylor Model Basin
RADM R.L. Moore, Jr.	Bureau of Ships
Mr. R.V. Morse	Sperry Gyroscope Company
Dr. Seizo Motora	University of Tokyo
Mr. John J. Nachtsheim	Bureau of Ships
Mr. B.V. Nakonechny	David Taylor Model Basin
Mr. Robert N. Newton	Admiralty Experiment Works
Mr. Thomas Noble	Sperry Piedmont Company
CAPT J.A. Obermeyer	Bureau of Ships
Dr. Kazuo Ochi	David Taylor Model Basin
Mr. C.R. Olson	David Taylor Model Basin
Mr. Frank Pavlik	Sun Shipbuilding and Dry Dock Company
CDR C.N. Payne	Bureau of Ships
Mr. Lennart Pehrsson	Aktiebolaget Karlstads Mekaniska Werkstad, Karlstads, Sweden
Dr. P.C. Pien	David Taylor Model Basin
Mr. D. Price	Sperry Gyroscope Company
Mr. Carl H. Renbein	Moore McCormack Lines, Inc.
Mr. L. Franklin Robertson, Jr.	Maritime Administration
Mr. Edward J. Rodgers	Ordnance Research Laboratory
Mr. Lester Rosenblatt	M. Rosenblatt and Son, Inc.
CAPT H.E. Saunders	Bureau of Ships
Dr. K.E. Schoenherr	David Taylor Model Basin
Mr. L. Segel	Cornell Aeronautical Laboratory
Mr. R.G. Seitz	Maryland Shipbuilding and Dry Dock Company
Dr. W.J. Sette	David Taylor Model Basin
Dr. Hisamitsu Shiba	Transportation Technical Research Institute, Tokyo, Japan
CDR J.W. Shultz	Bureau of Ships
Mr. A. Silverleaf	National Physical Laboratory, England
Dr. Bennett Silverstein	Bureau of Ships
Mr. Ivar D. Soelberg	Office of Naval Intelligence
Mr. T. Soo Hoo	Office of the Chief of Naval Operations
Mr. A.M. Stefano	Philadelphia Naval Shipyard
CAPT J.J. Stilwell	Bureau of Ships
Dr. A.G. Strandhagen	University of Notre Dame
Mr. J.S.C. Straszak	National Research Council, Canada
Mr. G.R. Stuntz, Jr.	David Taylor Model Basin
Mr. Anthony Suarez	Davidson Laboratory
Mr. Abraham Taplin	Bureau of Ships
Mr. R.J. Taylor	Maritime Administration
LT Claude R. Thompson	United States Coast Guard
Mr. Jan-Olof Traung	Food and Agriculture Organization of the United Nations, Rome, Italy
Mr. J. Vasta	Bureau of Ships
LCDR Elias Venning, Jr.	David Taylor Model Basin
CDR M. da C. Vincent	David Taylor Model Basin
Mr. Z.G. Wachnik	David Taylor Model Basin
LT J.W. Webster	United States Naval Academy
Mr. V.E. Williams	Sperry Piedmont Company
Mr. Harold L. Woods	Mare Island Naval Shipyard
Mr. C.L. Wright, Jr.	Boston Naval Shipyard
RADM E.A. Wright	David Taylor Model Basin
Mr. D.B. Young	David Taylor Model Basin
CDR R.J. Zoeller	David Taylor Model Basin

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INTRODUCTORY SESSION

The First Symposium on Ship Maneuverability was opened at 10 a.m. on 24 May 1960 at the David Taylor Model Basin, Washington, D.C. Mr. Morton Gertler, Chairman of the Symposium, opened the Introductory Session.

The following addresses were given at the Introductory Session.

INTRODUCTION

by

Rear Admiral E. A. Wright, USN
Commanding Officer and Director
David Taylor Model Basin

Mr. Chairman, Admiral James, Ladies and Gentlemen:

Noah's Ark was perhaps the only ship in which maneuverability was not an important characteristic. It seems almost incredible, therefore, that this is actually our first symposium devoted to exchanging ideas on ship maneuverability.

Even so, this one began with plans for a Second International Symposium on Seakeeping. Then in my office early last year, Dr. Schoenherr and Captain Saunders pointed out that we still have a profusion of undigested technical contributions and of unanswered questions from the First Seakeeping Symposium at Wageningen in 1957, and that we have much more to learn by pooling our ignorance and thoughts on maneuverability. And so here we are.

The timing seemed to fit nicely, between the American Towing Tank Conference held at the University of California in September 1959, and the International Towing Tank Conference planned for Paris in September 1960. The subject matter of maneuverability seemed an appropriate buildup to the 1960 ITTC at which, for the first time, a full technical session will be devoted to maneuverability.

The new maneuvering facilities here at the David Taylor Model Basin are still under construction. They will be opened for the first time during this Symposium, and you will have the opportunity to see them Wednesday afternoon. However, several months of shakedown, optimization, and correlation will be required before these facilities are ready for dedication and productive research.

To open the First Symposium on Ship Maneuverability, we are highly honored to have the Chief of the Bureau of Ships, Rear Admiral R. K. James. The David Taylor Model Basin is but one of the seven naval laboratories under the management and technical control of Admiral James, who is also responsible for our eleven shipyards and many supporting activities. The Chief of the Bureau of Ships is responsible for the design, construction, and maintenance of every ship in the United States Navy. The Bureau of Ships puts more effort into ship research than any other organization in the world. It is with particular pleasure that I introduce to you Admiral James.

WELCOMING ADDRESS

by

Rear Admiral R. K. James, USN
Chief, Bureau of Ships
U.S. Navy Department

The Bureau of Ships is interested, of course, in what effort goes into the field of ship research, but, even more important, we are extremely interested in what comes out of such effort. It is the hardware that is the responsibility of the Chief of the Bureau of Ships. Research leading to it is an important and major step, but the ultimate result is the finished hardware.

When I was a young naval officer, I had a rare experience that taught me the importance of maneuverability. I was undertaking the transit of a very restricted channel in an outboard driven boat where I had been doing some fishing off Ocean City, Maryland. As I entered the channel, I was almost overtaken by a following sea that was roaring up inside that narrow and restricted channel. If it were not for the complete maneuverability of that craft, I might not be participating in this excellent performance today. So I am indeed conscious of the importance, for many reasons, of the contributions that you as a group of International and American experts can produce in this field. It is most important that you do get together because the field is so limited. In this regard we can benefit by each other's mistakes and appreciate each other's advances in the programs that we are able to conduct collectively in the several basins around the world. So I commend you to your effort. This is a most vital one to us of the United States Navy, as it is to all who go to sea in ships. We have considerable interest not only in the applied research that is basic to the comment that I made in the beginning, but we are equally concerned with the fundamental research which goes on in this area.

I am aware that we put over 2 million dollars a year into the basic fundamental research in this field of ship maneuverability largely through the management and direction of the David Taylor Model Basin which also touches many dozens of universities and laboratories throughout our country. The composite effort that goes on in our country, added to that which is done in your country for those of you who have come from abroad, can really produce for us the kinds of things that we need; and maneuverability is certainly one of them.

I am asked to welcome you to this gathering, and it is indeed a pleasure, a distinct pleasure, to extend the welcome of the Navy Department and particularly of the Bureau of Ships to all of you who have come from your various offices and institutions around the world. I trust that you will have a most successful meeting. In exchanging ideas, I am sure you will produce for us those things that we need: the most maneuverable vessels that are possible for man's imagination to create. Good luck on your work and I hope we will get to see each other again soon.

SHIP MANEUVERABILITY FROM THE STANDPOINT OF THE NAVAL DESIGNER AND OPERATOR

by

Rear Admiral J. M. Farrin, USN
Assistant Chief
Bureau of Ships for Design,
Shipbuilding, and Fleet Maintenance
U.S. Navy Department

It is a particular pleasure for me to be with you today, as an old alumnus of the Model Basin, to take some part in this important first meeting on ship maneuverability. In these days of rapid technological advances, when we have seen such things as supersonic aircraft, nuclear power, and space rocketry, the more ancient art of naval architecture must not be left behind. I think our need today for important advances in the field of the naval architect and the marine engineer are, if anything, greater than ever before. As you know, developments in the field of submarine propulsion and maneuverability have made the submarine a much faster and a more agile vehicle than any of its predecessors. But, unfortunately, the surface ship has not kept pace with this. Now, if surface naval vessels are to maintain their competitive position relative to the submarine, it is necessary that they achieve a substantial increase in speed and become more maneuverable. This is a military requirement. Ship maneuverability then has to take on new dimensions in this era of rapid change. This change adds orders of magnitude to the problem which existed with previous conventional displacement ships at moderate speed.

Today we have hydrofoil boats, planing craft, semisubmerged high-speed vessels, and hover craft, and with these vehicles we are faced with an entirely new set of stability and control problems. But even in the slower-speed more-conventional ship, modern naval tactical evolutions require expert seamanship, and they require more precise ship control than has heretofore been obtainable. For example, the Navy performs important replenishment transfer-at-sea operations that are necessary to supply our combatant ships and prolong their endurance. Such operations require these ships to maneuver under varying sea conditions in close proximity to other ships for considerable periods of time. As a result of this requirement our recent designs of naval auxiliaries have stressed improved steering qualities.

This, of course, is not to say that our ability to design maneuverability into ships has not undergone considerable improvement in recent years. As a result of research we have attained some understanding of directional stability and its predictions. We have developed techniques of improving the maneuverability of our ships without resorting entirely to the trial and error methods. It has been some time, I'm glad to say, since we in the Bureau of Ships have been

faced with a directionally unstable ship, although I recall quite well some 12 years ago when we did have this problem with, of all things, the President's yacht WILLIAMSBURG. At the time we, of course, gave this problem high-priority high-level attention and came up with a solution I am sure is familiar to all naval architects. We simply increased the rudder area.

I mentioned earlier that replenishment at sea is a vital problem. There has been a continuing trend toward replenishment at higher speed. Adverse forces and moments have thus been greatly increased. Mr. Newton will survey this field for us in his paper this afternoon. But, at the present time, we cannot predict in advance the effect of hull form, lateral and longitudinal ship separation, speed, and rudder characteristics on the ability of ships to maintain these close operations. Also we have little basis to predict what to expect if these replenishment operations occur in rough seas. At present, avoidance of collision, unfortunately, rests almost entirely on the judgment of the ship captain. We think a scientific input and interest in this problem has been sorely needed.

Good maneuverability at low speeds, especially in restricted waters, as we all know, is very difficult to obtain. We know, for instance, that our capital ships and large merchant vessels require delicate handling in canals. Mr. Pehrsson will discuss bow propellers as a solution to this slow-speed maneuvering problem. Other devices that may help the problem are active rudders, jet flaps, and cycloidal propellers. Some problems of maneuvering in restricted waters will also be discussed by Mr. Bindel and Dr. Schoenherr.

But perhaps the most fundamental problem of all is just what do we mean by maneuverability. It is difficult indeed to ask the ship designer to build in satisfactory controls when we have not determined performance criteria for this. Messrs. Gertler, Gover, Segel, and Nomoto, in their papers, will, I'm sure, shed light on this important area. But, regardless of the absence of adequate criteria, the designer still has to provide his ship with a rudder. It is relatively easy to increase the power of the steering engine while the ship is still on the drawing board, but of course, once it is built, this capacity must remain fixed. The designer thus must have a reliable way of predicting rudder torque. Toward this end we have, of course, quite a fund of aerodynamic literature which we have extended to the lower aspect ratios that are associated with rudders. Mr. Taplin will review the present state of the art in this field. But even with this extensive background, we still have much to learn. What is the mechanism of the unwanted aeration of rudders? How does it affect maneuverability? Why do the existing tests of model and full-scale forces and moments show such poor correlation? What is the role of flow separation in this problem?

Through the ingenuity of our personnel and some excellent facilities, we have mastered many of the important problems of submarine maneuverability. For example, we can successfully satisfy the "opposing" requirements of directional stability and control in our submarines, as shown by the submarine SKIPJACK which can be flown hands-off and yet can maneuver almost as well as a porpoise does. We hope to achieve capability such as this with surface ships. Dr. Shiba will discuss some model tests along this line. And of course the new facilities that Admiral Wright has mentioned here at the David Taylor Model Basin should help answer this question.

Scientific probing into the fundamental hydromechanic problem is essential. Such probing, I think, is exemplified by Mr. Motora's paper on "Added Mass and Moment of Inertia." The designer needs the tools of both analysis and synthesis. Analysis will tell him the physics of what is happening and what the various components of the ship contribute toward the end result. Synthesis enables him to combine these various elements into a balanced, well-behaved ship. But the scientist has to provide the analytical information.

I am most happy to see the coincidence in time between the First Symposium on Ship Maneuverability and the opening, as Admiral Wright has mentioned, of the new Maneuvering and Sea Keeping Facilities here at the David Taylor Model Basin. I think in any discussion of maneuverability, we cannot help but be reminded of the important pioneering work of the late Dr. K. S. M. Davidson, since his early interest in the fundamental problems of stability, steering, and maneuverability contributed so much to the knowledge of our profession in these important problems.

The proceedings of this symposium will no doubt bring to light much new information on all aspects of the maneuvering problem. These should form important milestones in our progress. We in the Navy Department feel that the First Symposium on Ship Maneuverability is a most important event. We have high hopes for a very productive meeting.

SHIP MANEUVERABILITY FROM THE STANDPOINT OF THE MERCHANT MARINE DESIGNER AND OPERATOR

by

Ludwig C. Hoffman
Chief, Office of Ship Construction
U.S. Maritime Administration

The increasing size and speed of ships bring greater problems of safe operation in congested waters and control at high speeds in waves. Ship designs are a compromise with respect to maneuverability, first cost, and course-keeping ability. Obviously, a ship operator would like to have maximum maneuverability in port to minimize the cost of tugs and delays in docking the vessel. He needs a ship which will hold a steady course to avoid lost time and expense incident to unnecessary additional mileage when the vessel wanders off course or where excessive rudder angles are required to keep the ship on course.

Hull forms and appendages for conventional single-screw, general purpose, dry cargo ships have been stabilized over the years in the proving ground of actual service, with the result that generally acceptable performance characteristics are achieved as a compromise

with first cost and docking expense. This should not be construed as ruling out the possibility of improving maneuverability characteristics of these conventional ships without impairing other desirable characteristics. However, it appears that the most fruitful avenues of research can be found with ships destined for unusual services requiring superior maneuvering characteristics. Occasionally, and more often in the twin-screw category, we hear of conventional designs which have proven inferior from the standpoint of tactical diameter or course-keeping ability. These instances come to light usually after grounding or collision which turn out to be very expensive casualties. Modifications for improving the maneuvering characteristics of existing ships are, of course, many times more expensive than obviating the difficulty in the first place through use of the model basin tools now available.

The Maritime Administration is quick to encourage design agents and ship operators to fully exploit the facilities available in the tank for this purpose. We consider our construction subsidy participation to be a particularly wise investment when the engineering costs for preparing ship construction contract plans and specifications embrace a comprehensive program of model testing, including maneuvering and seakeeping aspects.

In the de ign and construction of the Coast and Geodetic Survey's ship, SS SURVEYOR, delivered earlier this month at the National Steel and Shipbuilding Company, San Diego, California, the Taylor Model Basin performed significant maneuvering experiments which gave a firm basis for selecting auxiliary propulsion. The Coast and Geodetic Survey requires many of the course-keeping abilities common to oceanographic ships, as is fully explained in Mr. Rosenblatt's excellent paper to be presented in a few days before the Society of Naval Architects and Marine Engineers. The experiments conducted with the limited facilities then available demonstrated to everyone's reasonable satisfaction that, for this particular service, the right-angle drive auxiliary propulsion was superior to an "active rudder." As a part of the Maritime research program, we are making available to the Coast and Geodetic Survey a right-angle drive auxiliary propulsion unit manufactured by Murray and Tregurtha which will be permanently installed on the SS SURVEYOR. The service experience of this installation will be made available to the profession and to the model basins where it is visualized that correlation testing not only in still water but in waves may be profitable to establish the degree of reliability which can be placed on such model testing.

With the opening of the St. Lawrence Seaway, a large group of cargo ships which were not designed for that particular service nevertheless were used in transiting the tortuous canals and locks of that system. One of the major elements of extra expense was in repairs resulting from damage to ship structure due to collision with the sides of the locks and the lack of control of the ships, particularly when they are light in restricted waters and subjected to strong crosswinds. This is one example where a capital investment in auxiliary maneuvering equipment would be fully justified.

Stimulated by Messrs. Gertler and Gover's paper presented before the Chesapeake Section of SNAME, the Maritime Administration included spiral maneuvers as part of the trial

agenda of the tanker AMERICAN EXPLORER completed last summer at Ingalls Shipbuilding Corporation. This is a 20-knot commercial type tanker, now being operated by the Military Sea Transportation Service, having rather fine lines for a tanker because of its high speed. The trials were conducted under adverse conditions of wind with 30- and 40-knot gusts blowing continuously. An analysis showed that the vessel has good course-keeping ability. Thus far the operating experience with MSTS tends to bear this out.

However, unlike the AMERICAN EXPLORER, there is an entirely different family of tankers under construction which inherently will give trouble from the standpoint of their course-keeping characteristics. Bethlehem Steel Company has recognized this in the case of the largest tanker in the world, 106,000 tons DWT, now under construction at Quincy, Massachusetts. At their request, the David Taylor Model Basin conducted extensive studies with various appendage configurations on this twin-screw design. Judging from experiments, the cost of the model tests will be repaid many times through improvements which were achieved in the behavior of the ship based on these tests. The Maritime Administration is insuring the mortgage of this ship design and is suggesting that spiral maneuvers be conducted, not only to provide the absolute answer on course-keeping ability, but to permit correlation between model tests and full-scale performance.

Summarizing from the standpoint of the marine designer and operator, there is a constant battle to decrease both operating and first costs in the economic struggle for profitable operation. A highly maneuverable ship can cut down docking fees and tug charges, have better access to "up river" ports, improve its safety through avoidance of collisions, and on occasions receive preferential treatment by harbor pilots so as to avoid delays required by the pilot to adjust trim conditions or wait for more favorable tides. When superior maneuverability can be obtained without prohibitive first cost and the ship operator is convinced of this fact, the improvements will be made.

The new facilities in the model basins such as those now being made available at the David Taylor Model Basin offer better opportunities to determine in advance of actual ship construction the gains which can be expected in the field of maneuverability and the relative cost of incorporating the desirable features in the vessels. The Department of the Navy is to be commended for its farsighted planning and action in this area.

THE ROLE OF THE RESEARCH AND DEVELOPMENT LABORATORY IN THE FIELD OF SHIP MANEUVERABILITY

by

Dr. Karl E. Schoenherr
Technical Director
Hydromechanics Laboratory
David Taylor Model Basin

The part assigned to me in this morning's survey of ship maneuverability is to discuss the role played by the research and development laboratory in advancing our knowledge in this field. To do this adequately would require considerably more time than has been allowed on the program for this discussion. Since time is of the essence, only two choices were open: either to cover the whole subject very sketchily, or to cover only a small part of it and do so in a reasonably adequate manner. The latter course appeared to be preferable. The part which I elected to discuss is a survey of the work done by the model basins on this side of the ocean, particularly EMB and TMB, on ship maneuverability within the past forty years. I want to make this limitation clear at the outset, as I do not wish our foreign friends and visitors to feel that we are unaware of or unappreciative of the excellent work along the same lines that has been done in other model basins in the world.

The subject of ship maneuverability is usually considered to include the determination of the motion of surface ships in a horizontal plane and the motion of submarines in horizontal and vertical planes under the action of the hydrodynamic forces applying on the hull and rudder, as well as the control of these motions. In this broad sense, therefore, maneuverability has two aspects: first, the ability of a ship to change course rapidly when course changes are desired, and, second, the ability for a ship to remain on course when no course changes are desired. In the past, greater emphasis has been placed on "turning ability" rather than on course-keeping ability, but in recent years both aspects of the subject have been receiving about equal attention. The earlier emphasis on ship turning ability probably stemmed from the fact that turning characteristics were relatively easy to determine and were important to know at that time when naval vessels still operated in compact squadrons rather than as single units, while course-keeping characteristics were not so easy to determine. The work done by the model basins reflects this attitude. Although steering and maneuvering tests were made occasionally in the early days, such tests were often made to correct bad steering conditions found on individual ships already built, or to evaluate patented rudders, such as the Oertz rudder and Contra rudder, and model turning tests were carried out quite regularly in the course of developing new vessels before construction.

The old United States Experimental Model Basin had no special facilities for turning and maneuvering tests. Whatever work was done there had to be done in the standard towing tank. Fortunately, this tank was wider than most contemporary model basins, having a width of 42.7 feet on the surface, which permitted turning of 12- to 20-foot models through about 90 degrees. In spite of these space limitations, turning tests were begun at EMB as soon as self-propulsion tests had proved feasible and reliable. This was fostered by the fact that at EMB all models were made of wood and thus could withstand considerable rough handling, and that from the start fully appendaged models driven by internal dynamometers were used instead of wax models with propeller pushed up from astern, as was the case at that time in European tanks.

It may be of interest to describe in some detail the technique for conducting turning tests developed at EMB as this technique greatly influenced subsequent development at TMB. The earliest tests were made in 1921–24 on a model of the battleship NEW MEXICO. Some of the results of these tests were reported in a paper by Hewins and Roop at the 1931 meeting of the Society of Naval Architects. However, for present purposes it will be of interest to repeat the salient point of this investigation quoting directly from the original EMB report. Report No. 106 dated 1921–24 states:

> "In 1918, experiments were made at sea on the NEW MEXICO, fitted with hydraulic steering gear, and attempts were made to determine rudder forces and moments while the vessel was turning under helm. These experiments were only partially successful, and the opinion was expressed that model experiments only could further clear the problem. Accordingly, the work of developing apparatus, which would give the desired results, was undertaken at the Model Basin. This work has been done under many difficulties, and it was not until 1924, that successful results were obtained.
>
> "The object to be obtained, in a model of the NEW MEXICO, self-propelled at a given speed and running free, was to turn, at a proper moment, the rudder to a predetermined angle, in a given time, and to furnish simultaneously continuous records of rudder force components, torque on the rudder stock, speed of vessel, curvature of path, helm angle, heel of vessel, and other minor data, all on a time base. All conditions of the experiments were to be as closely as possible similar to those of the trials of the full-sized vessel . . . .
>
> ". . . the width of this basin, about forty feet, precluded the use of a model longer than about ten feet. It involved small displacement, and small forces, the rudder pressure being of the order of one pound . . . .
>
> "The model was without bilge or docking keels, but was otherwise complete with four struts, shafting, and propellers to scale, . . ."

Deviating now from the report, I am showing you in three slides the model and instrumentation that was used. Figure 1 shows the exterior of the model, the mast erected at the forward end, and the camera platform directly over the rudder apparatus. Figure 2 shows the interior, the batteries and propulsion motors, and four small electric lights—one mounted at the foot of the mast, one at the masthead, and two on the centerline of the model forward of the rudder. These lights served to fix the position of the model while turning. Figure 3 shows the rudder apparatus. This apparatus consists of a bedplate fastened to the deck of model, on which rests a floating platform carrying the rudder and the rudder-drive mechanism. The two platforms are connected by cylindrical springs in such a way that the upper platform can move relative to the fixed bedplate with two degrees of freedom. The relative motion of the platform is measured by four Ames gages, the deflections of which were recorded by means of a moving-picture camera mounted above it.

Quoting further from the report:

> "To obtain the [turning] path of the model in the horizontal plane, and other data, an 8 × 10 plate camera, with axis vertical, was secured to the roof structure of the building, about 20 feet above the water. A disc shutter was operated at constant speed by a motor, the plate being thus given short exposures at constant intervals of about one-half second, and thus a record was obtained on the same plate, of the successive positions of the various points of the model traveling under it. These points were small electric lights, one at a forward masthead, one immediately below it, another aft on deck, and a fourth, also aft, which was illuminated with the starting of the steering motor, . . ."

Runs were made as follows:

> "The model was guided by hand, alongside a platform fixed to the basin carriage, making a straight run with the carriage until the point was reached where the helm was to be put over. The propelling machinery was operated during this run, regulated to give the desired speed. When the proper point was reached, the lights were turned on, together with the recording camera, and finally the steering gear switch was thrown on, and the model instantly released. As it thus proceeded free, the carriage was maneuvered to avoid it. At the same time, the vertical camera was operated."

Regarding the results obtained on this model and comparison with measurements on the ship, the report states:

> "The above agreements seem to make further comments unnecessary. It seems safe to assume that model results will give reliable values for the ship."

As previously mentioned, the technique and instrumentation developed for the NEW MEXICO model tests were the basis for subsequent work at EMB and TMB. Subsequently, free-running turning tests were made with larger models in which the turning path and heel

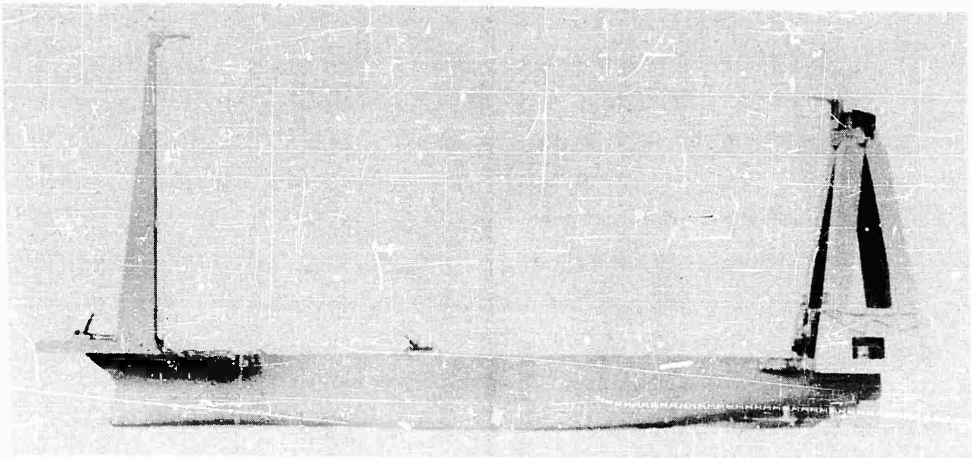

Figure 1 – Elevation of USS NEW MEXICO Model C & R Dept., Navy Yard, Wash., D.C. 1924

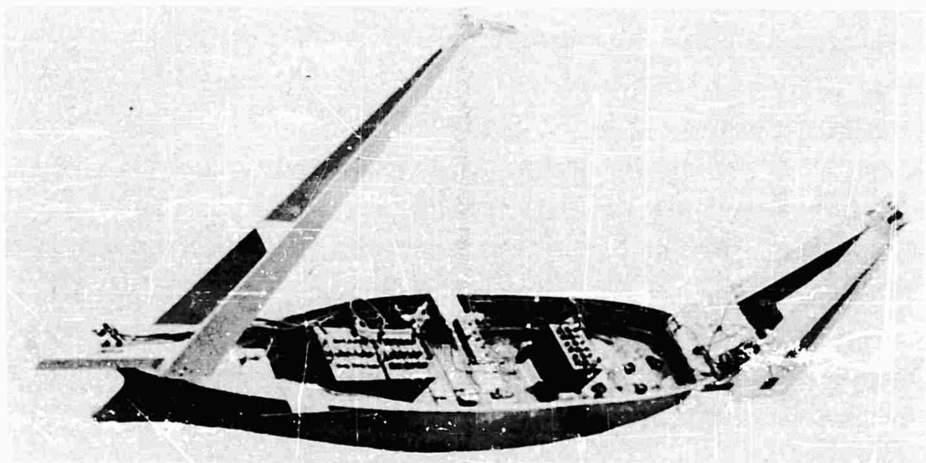

Figure 2 – Plan of USS NEW MEXICO Model C & R Dept., Navy Yard, Wash., D.C. 1924

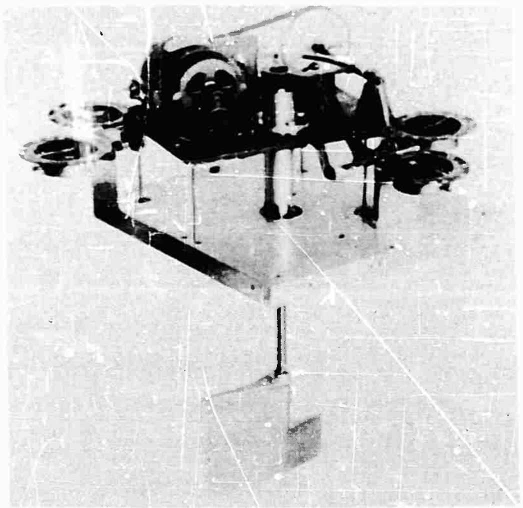

Figure 3 – Rudder Dynamometer

angle but no rudder forces were obtained. These larger models were supplied with power from the towing carriage through a cable that was dangling from a "fish-pole" vertically over the model so as not to restrain its motion. Verticality of the cable was maintained by allowing the carriage to follow the model and swinging the pole in azimuth. The path of the model was recorded as previously described, except that two synchronized cameras were used to cover the model approach as well as the turn, instead of the one camera used in the NEW MEXICO tests. As stated, rudder forces in the turn were not measured, but these were approximated by special straightaway runs in which the model was restrained from turning, and the side force on the stern of the model and the stock torque were measured. While the limitation of this procedure was realized, it soon became standard procedure on account of its simplicity and because the forces measured represented maximum values which are of primary interest to the designers of rudder and rudder engine.

As previously mentioned, the methods developed at EMB greatly influenced the design of the J-Basin at TMB. This basin, through which you will be conducted in the course of your visit, permits the turning of 20-foot models through about 180 degrees. The techniques of conducting tests in this basin are essentially the same as described in my previous discussion except for refinements of operation and instrumentation. Thus, the path of the model is still recorded by overhead cameras, but timing by means of flashing the lights on the models in regular intervals has superseded the timing by rotating shutters. The measurement of rudder forces is still done by spring deflection, but recorded by highly refined strain-gage equipment. And, last but not least, the old "fishing pole" has become a crane-like structure called a "rotating chair" pivoted at one corner of the carriage.

Figure 4 shows this equipment. To the left is the rotating chair carrying one member of the test crew who maintains the cable connecting carriage and model in a vertical position, and in the foreground is the model executing a turn.

I mentioned earlier that model steering and maneuvering tests were carried out at TMB in isolated cases. Thus, tests were run to measure the effect of rudder movement on shaft horsepower, tests to determine the ability of river towboats to control long trains of barges, tests to observe the yawing motion of barges towed by a towboat in waves, and many more. However, maneuvering tests of the Kempf type were not carried out before 1940, and tests of the Dieudonné type, now conducted at TMB as standard procedure, were not made before 1950.

In the early part of the war, the problem of maneuverability received increased attention which led to developments in two directions. First, plans were made to construct at TMB a rotating arm and an X-Y Basin. The rotating arm was to be a relatively simple outdoor affair but the X-Y Basin was to be an immense indoor facility. This basin was to be rectangular with wavemakers at one end, spanned by a carriage traveling on rails in the direction of the long axis of the rectangle, and equipped with a second carriage or cab suspended from the main carriage traveling in the direction of the short axis of the rectangle. The plans for these facilities were actively pursued, but when it became apparent that they could not be completed in time to solve pressing wartime problems, work on them was suspended, to be

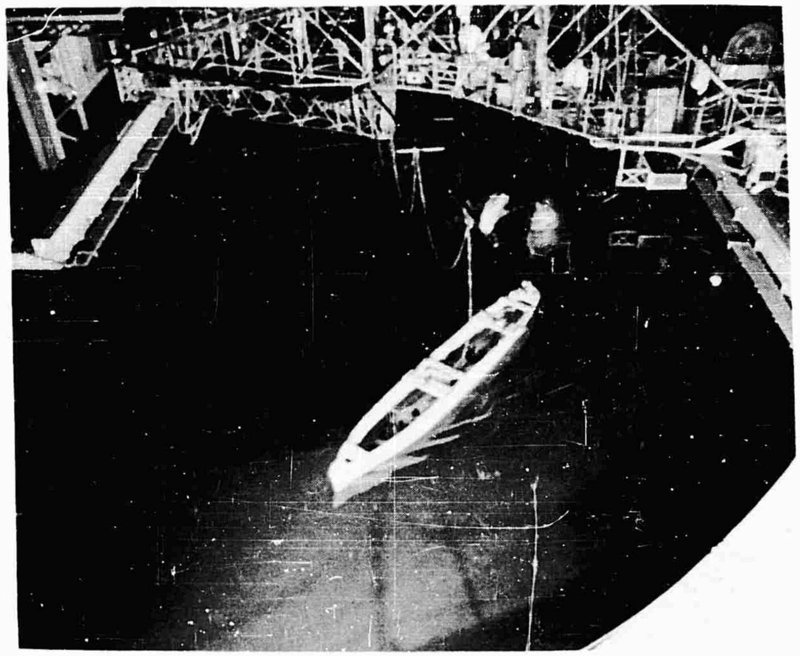

Figure 4- Turning Test with a Self-Propelled Model

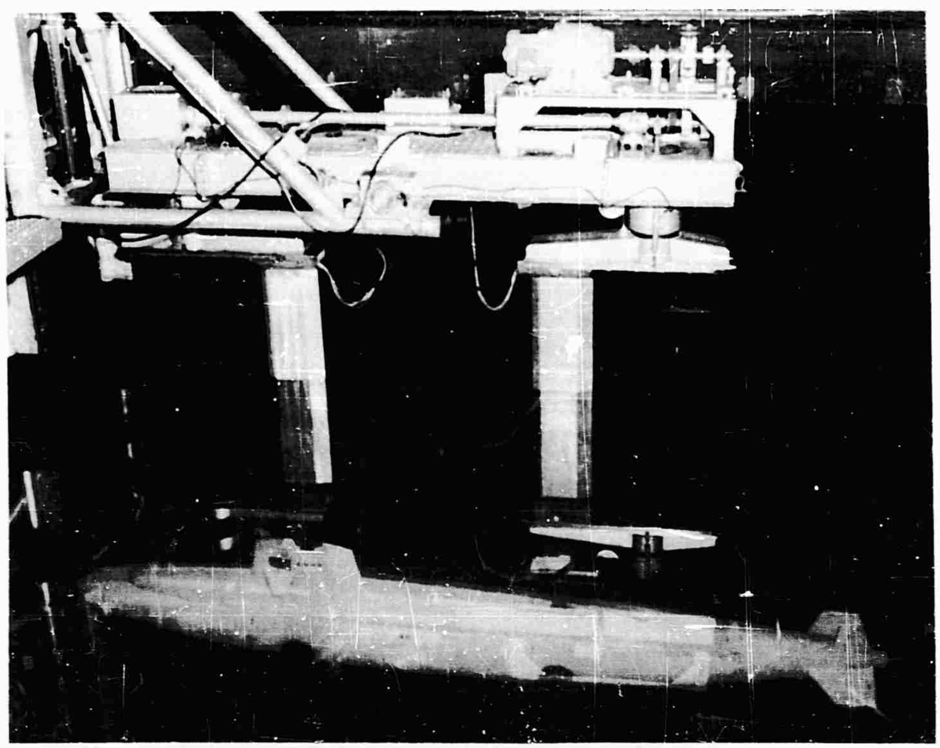

Figure 5 – DTMB Planar-Motion-Mechanism

resumed later. The new Maneuvering and Seakeeping Facilities, which will be demonstrated tomorrow, are the direct sequel of these earlier plans.

The second development I mentioned was the active support by the Navy of the Stevens Institute. Stevens, with Navy funds and encouragement, had constructed a 75- by 75-foot maneuvering basin around 1940. Early tests had shown that useful results could be obtained with 5-foot models used there. This, and the preoccupation of TMB with other work, brought about that a large part of the turning and maneuvering work that normally would have been done at TMB was transferred to Stevens. Close liaison was maintained between the two laboratories so that it is not unfair to say that at that time Stevens was practically an arm of TMB. After the war this close liaison ceased and TMB resumed maneuvering and turning work, proceeding with the development of suitable facilities that was interrupted earlier.

Up to now my discussion has been confined to surface vessels. Obviously, turning ability as well as course-keeping ability are of equal or even greater importance for submarines, torpedoes, and other self-propelled underwater vehicles. At the Experimental Model Basin, static tests with submarine models 12–15 feet in length were carried out quite regularly from about 1920 on. In these tests, the model was towed as deeply submerged as possible at various angles to the horizontal with and without diving planes, and lift, drag, and moments were measured. These tests enabled prediction of the effectiveness of various control surfaces, and no doubt contributed materially to the development of the modern U. S. submarine.

When activities were transferred from EMB to TMB these tests were continued, but other methods that would enable prediction of the dynamic behavior of underwater vehicles, as well as the behavior under steady-state conditions, were looked into. It was fully realized in those days that the techniques developed by aeronautical engineers for determining the dynamic stability of airplanes could be transferred to submarine work with no essential change and that the coefficients in the differential equations of motion for small departures from equilibrium conditions could be obtained by forced oscillation techniques. However, instrumentation to apply this technique did not progress beyond the sketch stage then and it remained for the younger generation, notably Messrs. Gertler and Goodman, to develop this instrumentation as well as the techniques for obtaining the coefficients and solving the equations of motion.

The heart of the instrumentation is the so-called Planar Motion Mechanism. This mechanism was briefly described in Admiral Wright's paper read before the Society of Naval Architects in 1958, and in Mr. Gertler's paper presented at the 1959 dedication meeting of the new model basin in Zagreb, Jugoslavia. To make this talk complete, I am showing you the mechanism in Figure 5. It consists of a table attached to Carriage II, and two long struts connecting the model to the table. The table is pivoted, which permits setting the model to any desired trim angle. In addition, the struts may be moved up and down relative to the table in-phase or out-of-phase, which permits inducing in the model a heave and pitching motion while advancing at steady or variable speed. The hydrodynamic forces on the model are measured by variable reluctance gages interposed between the model and struts.

Recording and resolution of forces into components is accomplished by a very complex electronic apparatus. The Planar Motion Mechanism is also used for yawing tests of submarines, by merely rotating the model about its longitudinal axis through an angle of 90 degrees.

The perfection of instrumentation in the past ten years has made it possible to perform now also free-flight turning tests on submerged submarines. In such a test, shown by Figure 6, the model is brought up to speed on a straight course by its propellers and kept on course and at a fixed depth by manipulating the rudder and diving planes from the carriage, which is following, and supplying the power. At a given instant, the rudder is thrown over to a fixed angle and held there while an attempt is made to maintain the initial depth. The roll and heel angles of the model throughout the maneuver are recorded by a gyro mounted in the model.

My talk would be incomplete without reference to the state of present knowledge of the subject of maneuverability of ships and what remains to be done in the future.

First, I should like to show to you three charts which demonstrate the increased interest in maneuverability as measured by the amount of attention it is receiving.

The first, Figure 7, shows the relative number of free-running model tests conducted through the years at EMB and TMB. You will note that the bar graphs are broken down into sections, each of which shows the type of information to be obtained.

Figure 8 shows much the same information but pertaining to captive model tests.

Previously, I have not said anything about full-scale testing because my discussion is primarily concerned with the role of the research laboratory in the field of maneuverability. However, EMB and TMB, like all model basins, have been in close contact with full-scale work throughout their existence—first as observers on standardization trials, then as participants, and lately as principals in conducting special research trials.

Figure 9 shows that full-scale testing is increasing at about the same rate as model testing. This, I believe, does not stem from distrust of the model results but from a natural inclination of the engineer to take no chances. Ships are becoming larger and ever more costly. Hence, the expense of a proof-test at the beginning of the life of a ship is a small premium to pay for the assurance that anticipated results are obtained.

As a result of the increased attention devoted to maneuverability, as shown by the last three figures, much has been learned but a good deal remains to be learned. As background, let me quote a statement I made at a meeting of Section 12.1 of NDRC held at the Stevens Institute in March 1944. I stated:

> "The value of Admiral Taylor's Standard Series lies in its identifying the main variables of ship resistance. We have no such data for turning but we should have something of the sort; that is, information concerning the principal factors which influence turning and maneuvering perhaps in the form of charts, so that it will be possible to predict turning performance for a new design. The preparation of such data will involve a detailed study of the

forces on the hull while turning under various conditions perhaps by local pressure measurements and by measurements of lateral forces on models traveling in circles (that is by rotating arm techniques)."

Since that time, the Stability and Control Division of the Laboratory, under Mr. Gertler's leadership, has devised for submarines something of the sort I had in mind. However, information of this kind for surface ships is still lacking. Hence, as I see it, one of the principal objectives of near-future investigations in the field of maneuverability should be the determination of the factors that make one surface ship maneuver well and another one poorly; in other words, to find the connecting link between hull form characteristics and turning and course-keeping ability.

The model test technique guided by theory is the ideal medium to obtain the answers. Statistical evaluation of full-scale trials is helpful, but progress by such methods is slow and costly. On the other hand, in the laboratory a very large number of variations can be tried, one variation at a time, at relatively low cost, and the test conditions can be carefully controlled. By applying this method in the superb new facilities now nearing completion I am confident that useful results will soon be obtained.

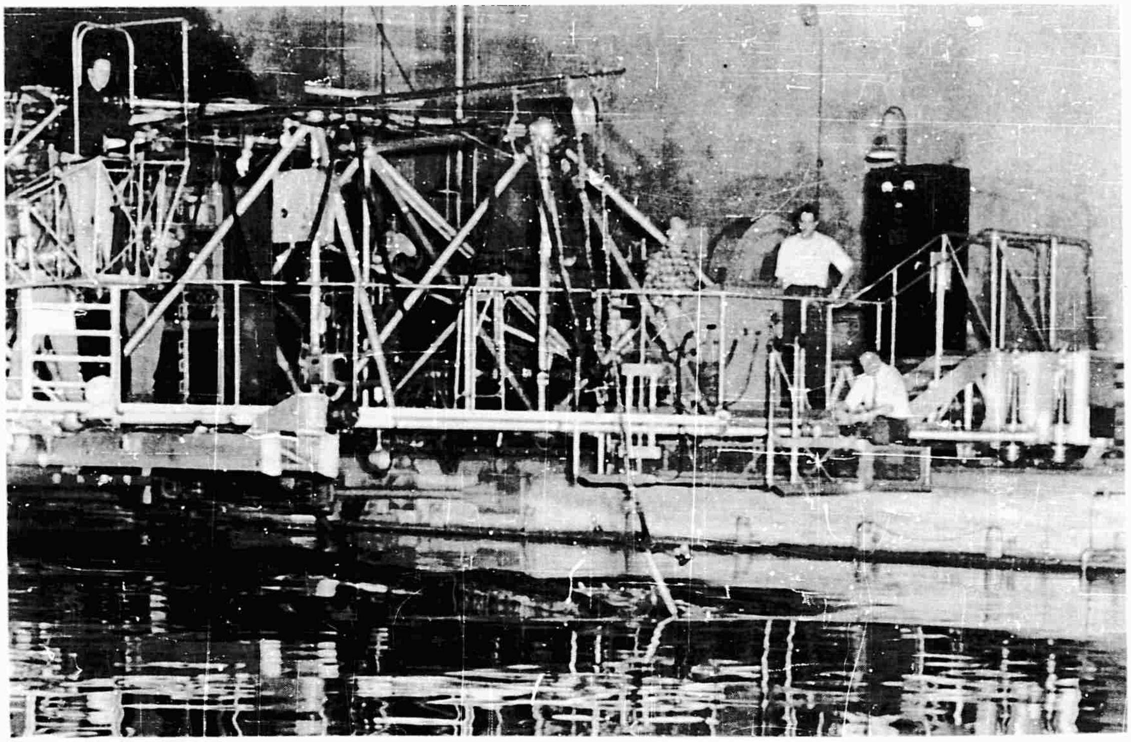

Figure 6 – Submarine Model Undergoing Submerged Turning Test

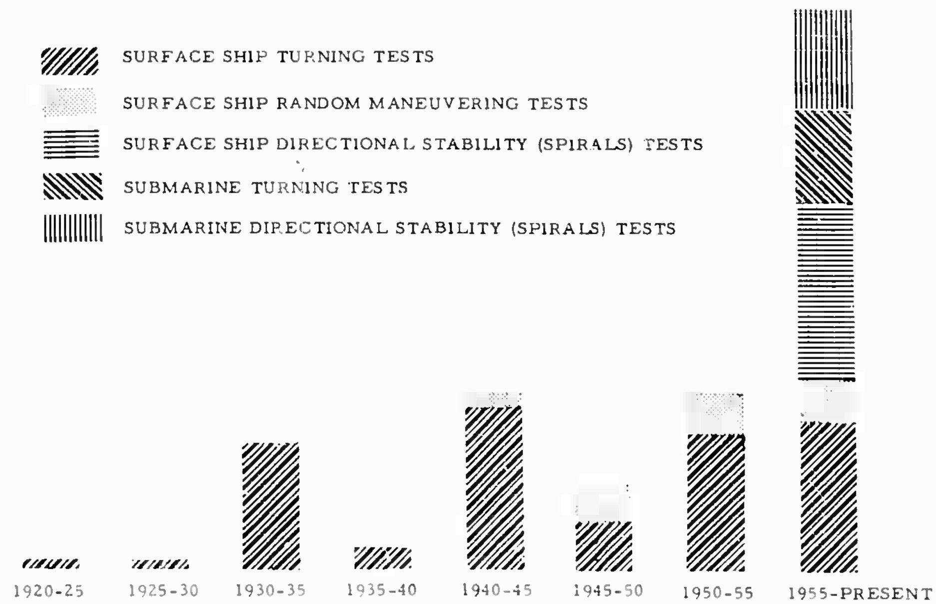

Figure 7 – Relative Amounts and Kinds of Free-Running Model Tests Conducted at EMB and TMB

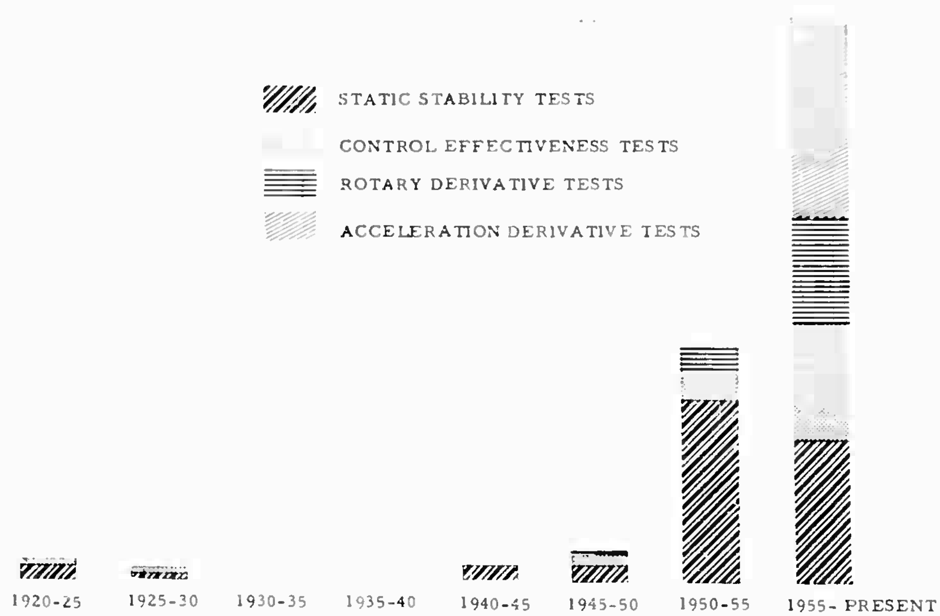

Figure 8 – Relative Amounts and Kinds of Captive Model Experiments Conducted at EMB and TMB

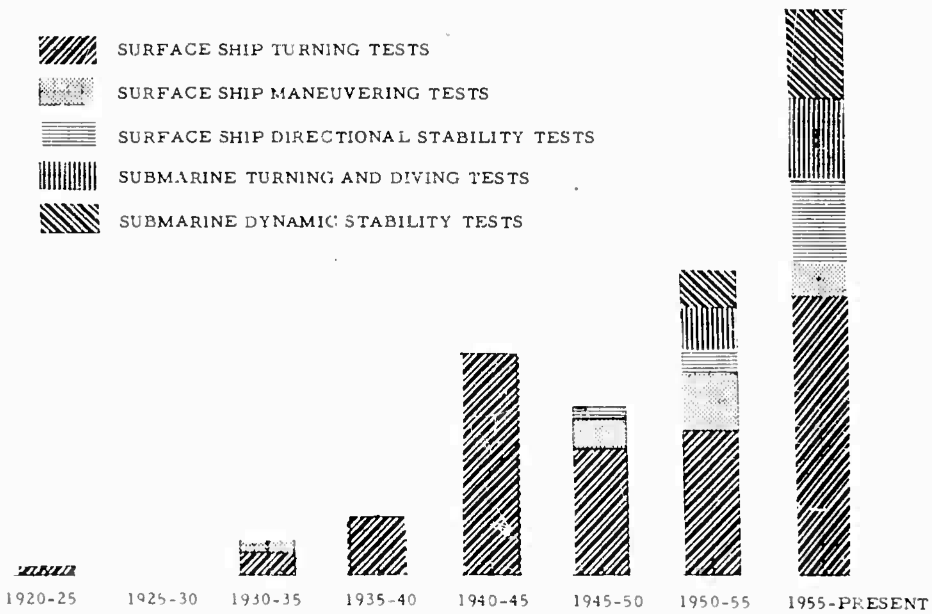

Figure 9 – Relative Amounts and Kinds of Full-Scale Trials Conducted by EMB and TMB

CLOSING REMARKS

by

R. N. Newton (R.C.N.C.)
Superintendent
Admiralty Experiment Works
Haslar, England

If Admiral Wright will excuse me, I am going to take advantage of almost the last two words in the program called "Closing Remarks." I do not know whether I was supposed to make these or not but I am certainly going to take advantage on behalf of the foreign delegates.

The subjects we have been discussing are approximately 70 years old, technically and scientifically speaking, anyway. If you study the bibliography, and incidentally one can do no better than to look at this publication by Nils Norrbin, of the Swedish State Tank, in which he summarizes the whole history right back to 70 years ago, and you will find there have been about 80 authors on the subject. In the whole of that time, they have written only about 100

papers, including the papers that have been given at this symposium. The point about that is that two-thirds of these papers have been given in the last 15 years. I don't think that can leave any doubt as to the importance with which I regard the subject, and the need for international cooperation for improving the knowledge.

It is in this concept that this symposium on the subject plays a highly important initial role in spotlighting international interest in no uncertain manner. The value to be placed upon this international get-together is matched only by the quality and variety of the papers, and also incidentally by those additional papers by authors who have not been given the opportunity to present them. The quality of the oral discussion has been equally high; it seems to have reached a crescendo.

If you need any further proof of the international interest in this subject, then I would recommend you to the names of the delegates who have been attending. And, of course, if one needs further proof of the interest displayed in the subject, then study Admiral Wright's paper itself where he discusses or describes so many new facilities being brought into use. (See E.A. Wright, "Some International Aspects of Ship Model Research," Journal of the American Society of Naval Engineers, February 1958.)

Now I said that as an excuse really, because in the light of that, on behalf of the foreign delegates here, may I have the utmost pleasure and sincerity in conveying to Admiral Wright, the Commanding Officer and Director here at TMB, and to those of his staff upon whom the burden of organization has fallen (and we all know how good that has been) the gratitude and admiration of all the foreign delegates and of the countries that they represent for their inspiration and efforts on our behalf and not to forget the excellence of the social part of the program.

Just to wrap it up, perhaps I might be permitted to coin a phrase which I rather feel reflects the spirit with which this symposium has been sponsored and conducted. It is very simple: "Research is a process which, properly controlled and amicably conducted without relevance to creed or politics, brings lasting benefit to all and disrepute to none."

SOME NOTES ON INTERACTION EFFECTS BETWEEN SHIPS CLOSE ABOARD IN DEEP WATER

by

R. N. Newton (R.C.N.C.)

Superintendent
Admiralty Experiment Works
Haslar, England

ABSTRACT

In this paper the author summarizes some model experiments and full-scale trials carried out to determine the feasibility of the operation of replenishment in deep water. The magnitude and sense of the interaction force and moment on the models are plotted in relation to their longitudinal and lateral separation, and interpreted to establish the sequence of corrective rudder movements, as one vessel overtakes another, to maintain parallel courses. The mean rudder angles estimated from model experiments are correlated to those used on ship trials.

Although the data obtained from these investigations is limited in scope, some interesting inferences are drawn as to the nature of the corrective action to be taken to avoid collision when two ships find themselves in close proximity on parallel or somewhat converging courses.

1. INTRODUCTION

The classic and original work on the reactions of vessels under way and in close proximity to one another was the investigation carried out by the late Rear Admiral D.W. Taylor, USN, the results of which he presented to the Society of Naval Architects and Marine Engineers (SNAME) in June 1909 in a paper entitled "Some Model Experiments on Suction of Vessels."[1] Whilst the discussion on this paper was rather dissentient, as might be expected on a subject not only new and complex but also not without its legal aspects there was general agreement on the need for further investigations on similar lines.

It is therefore rather surprising that in a brief survey of available bibliography on interaction effects in ship maneuvering the author has uncovered very little work to endorse or supplement Taylor's work, although there is a wealth of data on investigations into such effects in restricted waters and canals. The latter has arisen, no doubt, from the greater difficulties of navigation under these conditions, and yet when collisions in open deep water occur, as they still unfortunately do in spite of modern navigational aids, the question immediately arises as to what part interaction effects may have played in the accident. Notable examples which have been the subject matter of papers presented to technical institutions include the OLYMPIC and HAWKE,[2] and the QUEEN MARY and CURACOA.[3] Similarly some "near misses" have been recorded, including that of the cruiser EURYALUS and destroyer WORCESTER,[3] and the tugboat SVAVA and suction dredge ROLF.[3] The latter case is significant in that it emphasized the fact that in confined waters, although collision may be avoided, the event may result in one of the vessels grounding or ramming another vessel or pier in the vicinity.

[1]References are listed on page 20.

A most notable theoretical contribution on the subject is described in an excellent thesis by Silverstein[4] in which solutions are shown to be possible for mathematically shaped submerged bodies on the usual assumptions of a perfect fluid and linearization of the boundary conditions of the surfaces of the models. The interesting fact emerging from this theoretical study and that of Havelock,[5] is that, in spite of the limitations imposed by such assumptions, the theoretical curves derived for the interaction forces and couples are remarkably similar in character to those obtained by Taylor and in the experiments described in this present paper.

It is, however, with some diffidence that the author presents the results of experiments carried out between 1946 and 1948 at Admiralty Experiment Works (AEW), Haslar, where the library of publications by other authorities is by no means complete. Consequently he is conscious of the fact that the Haslar experiments may have been outmoded by more modern ones, in which case he can only express the hope that the results presented may endorse those obtained by other establishments.

The investigations at AEW were inspired by a military requirement, in this case the replenishment of warships with fuel from tankers, a procedure which is now regarded as commonplace and accompanied by little risk, once the vessels have taken up station close aboard. It is the process of taking up or breaking away from the abeam, or "fueling" position, that presents navigational risks and which this paper attempts to illustrate.

Two series of model experiments were carried out, the first with constrained models, one of the battleship H.M.S. KING GEORGE V and the other of the R.F.A. OLNA, to measure the forces and turning moments on each. In the second series the same models were freely propelled and controlled from the experiment carriage to study their behavior as one overtook and broke away from the other, and to record the rudder movements of each.

In addition, trials at sea were carried out to determine the ability of different classes of warships to take station abreast of a replenishment ship of similar size and form to the OLNA.

Before describing these experiments and trials, a brief explanation of the cause of interaction between ships in close proximity would seem to be indicated. Figure 1 shows the general character of the pressure field in the vicinity of a ship moving in open water due to the velocity distribution around the hull. Pressure changes of measurable amount occur at considerable distance from the hull. Any interference which modifies the pressure field necessarily has its reaction on the forces acting on the ship in greater or less measure. A well-known example is the change of sinkage and trim which a ship experiences in moving into shallow water. The change in the pressure field of the ship is accompanied by a change in buoyancy distribution which is manifested by a change in sinkage and trim.

Similarly, when two ships pass close by on parallel courses the pressure fields mix and the effect is to produce an unbalanced force and couple on each ship which must be counteracted by the rudder for each to maintain course or avoid collision. The magnitudes of the forces and couples, and their sense, varies according to the relative size and form of the ships, their relative headings, influence of propellers, and depth of water. The ability of either ship

to take corrective action depends also upon her maneuvering qualities and, in particular, obviously her response to rudder. Overall, the subject is complicated by the state of sea and wind.

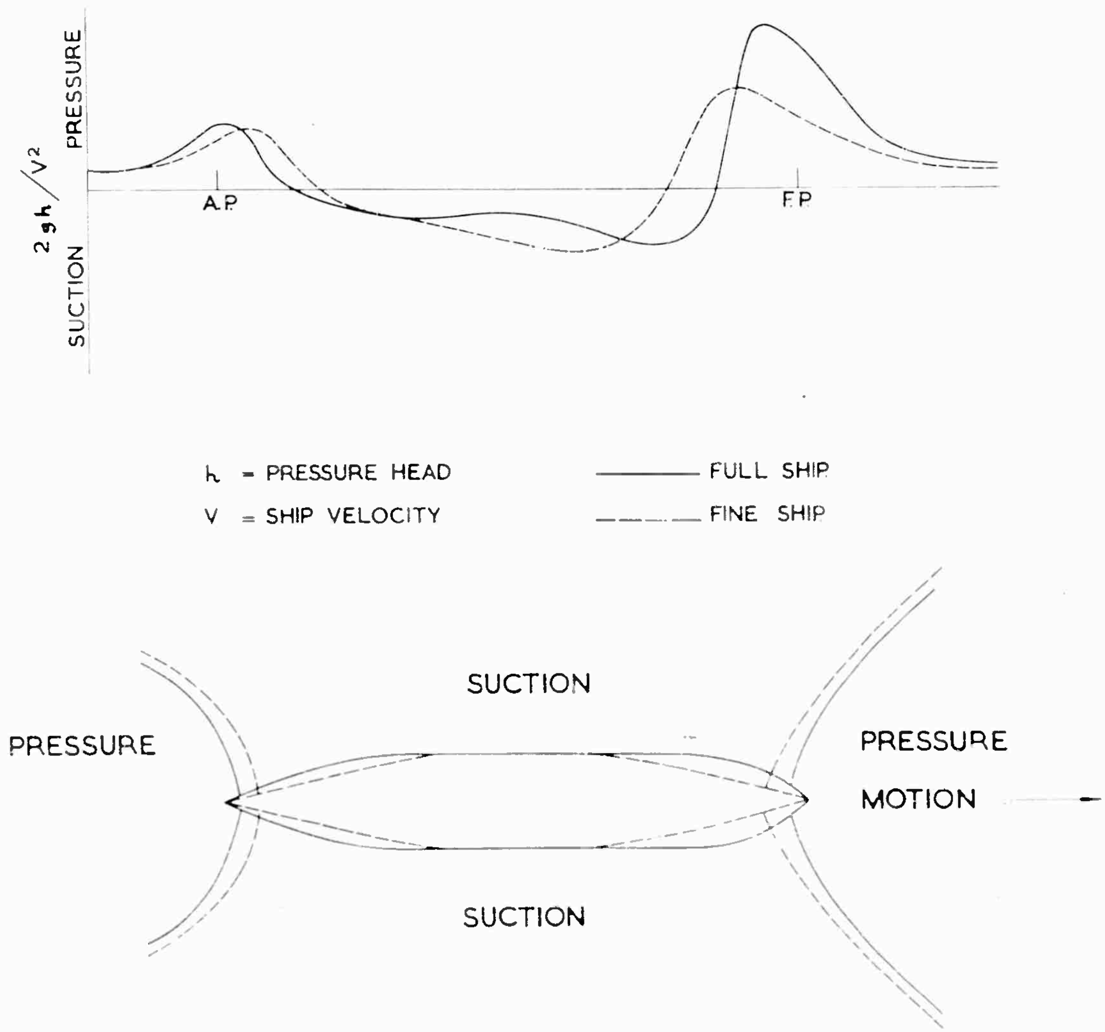

Figure 1 – Pressure Field Created by a Ship in Open Water

It is necessary to emphasize that the contents of this paper are mainly limited to the effects on fine form models or ships, of one size relation, on parallel courses in deep water, and in calm weather conditions.

2. EXPERIMENTS WITH CONSTRAINED MODELS

Models of the KING GEORGE V, designated Ship A, and the OLNA, designated Ship B, were made to 1/50 scale. Ship particulars represented by the models are given in Table 1.

TABLE 1

Item	Ship A	Ship B
Length on W.L., ft	740	567
Beam, ft	103	70
Draft, ft	29.3	30
Displacement, tons	36,890	23,570
Block coefficient	0.611	0.714
Corresponding depth of water	75 fathoms	

The models were towed on parallel courses at different positions relative to each other longitudinally, over a range of corresponding speeds from 10 to 20 knots, and at two separations, 50 and 100 ft, beam-to-beam transversely. The models were allowed complete freedom vertically, and fitted with rudders set amidships but without propellers. One model was towed on the resistance dynamometer on the middle line of the carriage and constrained on a straight course by lateral force guiders (as normally used for determining initial ship moments in rudder experiments) attached near each end of the model. The other model was towed by a rigid connection on a parallel course by guiders which prevented yaw but allowed vertical freedom. This procedure was then repeated to reverse hand, measuring on the other model. Additional experiments were made in the abeam position for various separations from 25- to 150-ft beam-to-beam. In all, the series of experiments involved about 1500 runs.

The forces measured at each lateral force guider, i.e., at each end of the model, have been converted to a single force acting at the "neutral point" positioned 0.2L from the bow and a couple, or turning moment, about this point. The reason for this is explained later. It should be noted that in Figure 9 of Reference 6, the forces were converted to the c.g. position. The forces and turning moments thus determined are plotted nondimensionally for 50-ft and 100-ft separation in Figures 2 and 3, respectively, to a base of longitudinal separation between the amidship points as A overtakes B from 600 ft astern to 600 ft ahead.

Referring to the lateral force curves, as the bow of A begins to overlap the stern of B, a force of repulsion develops on each and increases to a maximum when the bow of A comes abreast the midship point of B, becomes zero as the bow of A comes abreast of the bow of B, and changes sign to a force of attraction which reaches a maximum soon after the ships come abreast. As A begins to pass B, the forces undergo similar changes but opposite in sense. The greatest force is one of attraction arising from the addition of the ships' suction fields as they come abreast. Similar forces act on B, in the same sense and nearly in phase with those on A. It is worth noting the actual magnitude of these forces of attraction – about 26 tons on A and 35 tons on B when they are moving 50 ft apart at 10 knots, and four times as large at 20 knots. The forces are nearly halved when the separation is doubled to 100 ft.

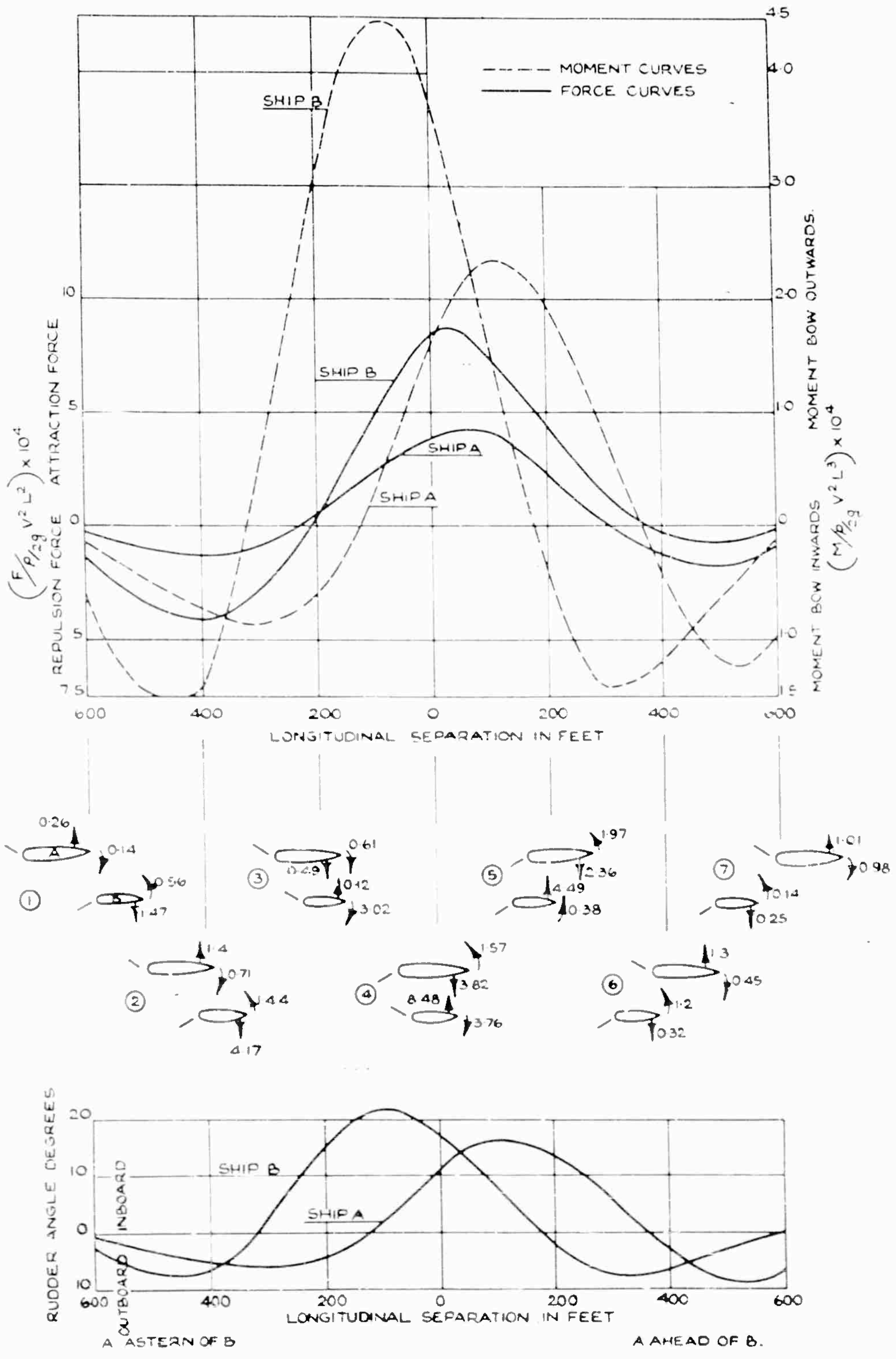

Figure 2 – Measured Interaction Forces and Moments and Correcting Rudder Angles

Fifty-foot separation beam-to-beam.

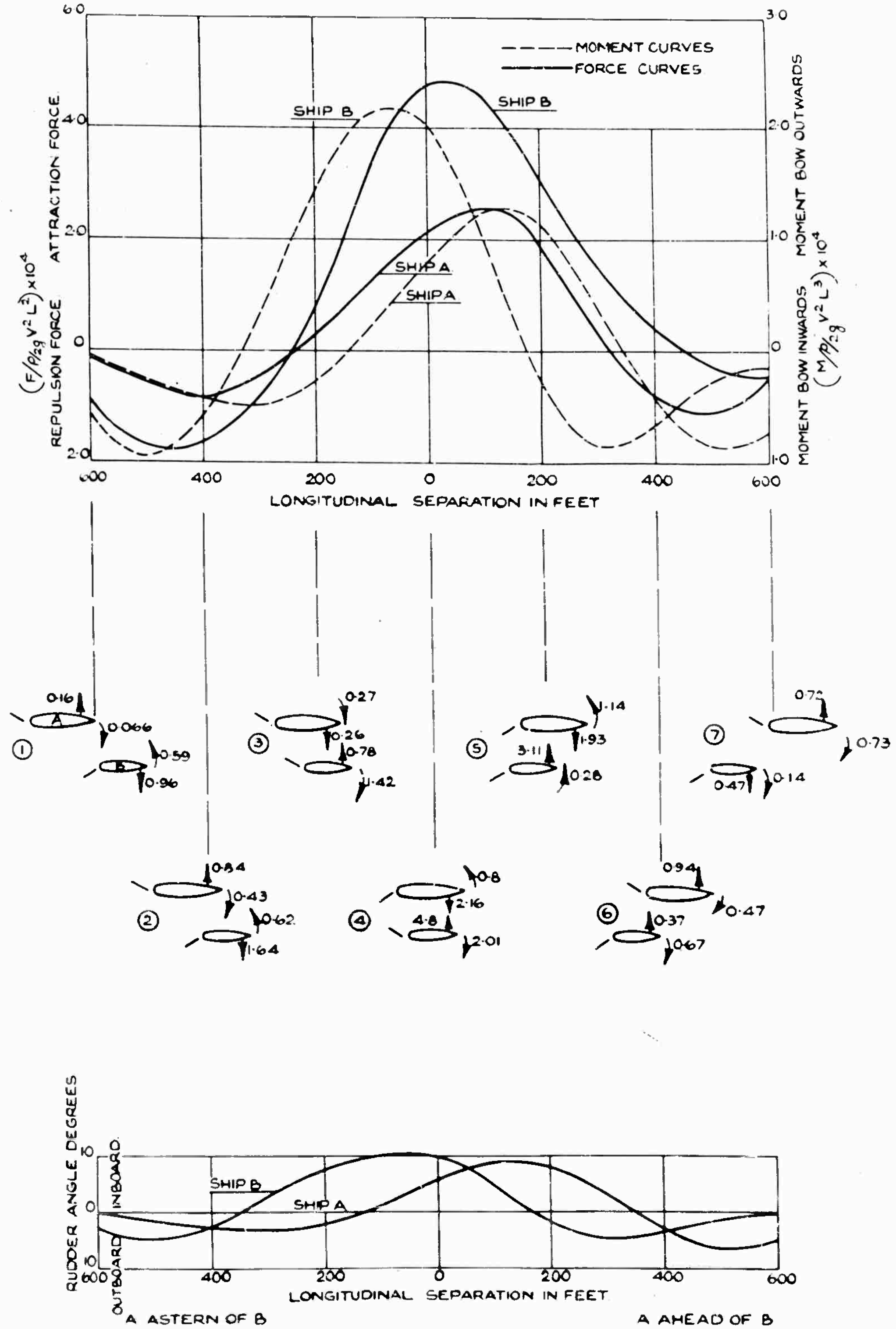

Figure 3 – Measured Interaction Forces and Moments and Correcting Rudder Angles

One hundred-foot separation beam-to-beam

The turning-moment curves follow a similar oscillatory trend. Before A begins to overtake B, she experiences a small bow-inward turning moment which at first increases and then decreases to become zero just as her bow comes abreast of the bow of B, and then changes to an outward turning moment which reaches peak value just as their sterns come abreast. From here on, the outward turning moment decreases to zero as the stern of A passes the amidship point of B and changes to an inward moment which is still quite appreciable when the ships are quite clear of each other. The turning moments on B follow the same trend, but lag behind those of A.

It is shown later that it is impossible to keep the ships' heads on course as in the model experiments by applying correcting rudder, and that to achieve simultaneous balance of interaction force and couple, the ships must yaw slightly.

The results of the experiments at various separations between 25 and 150 ft for the abeam position only are given in Figure 4.

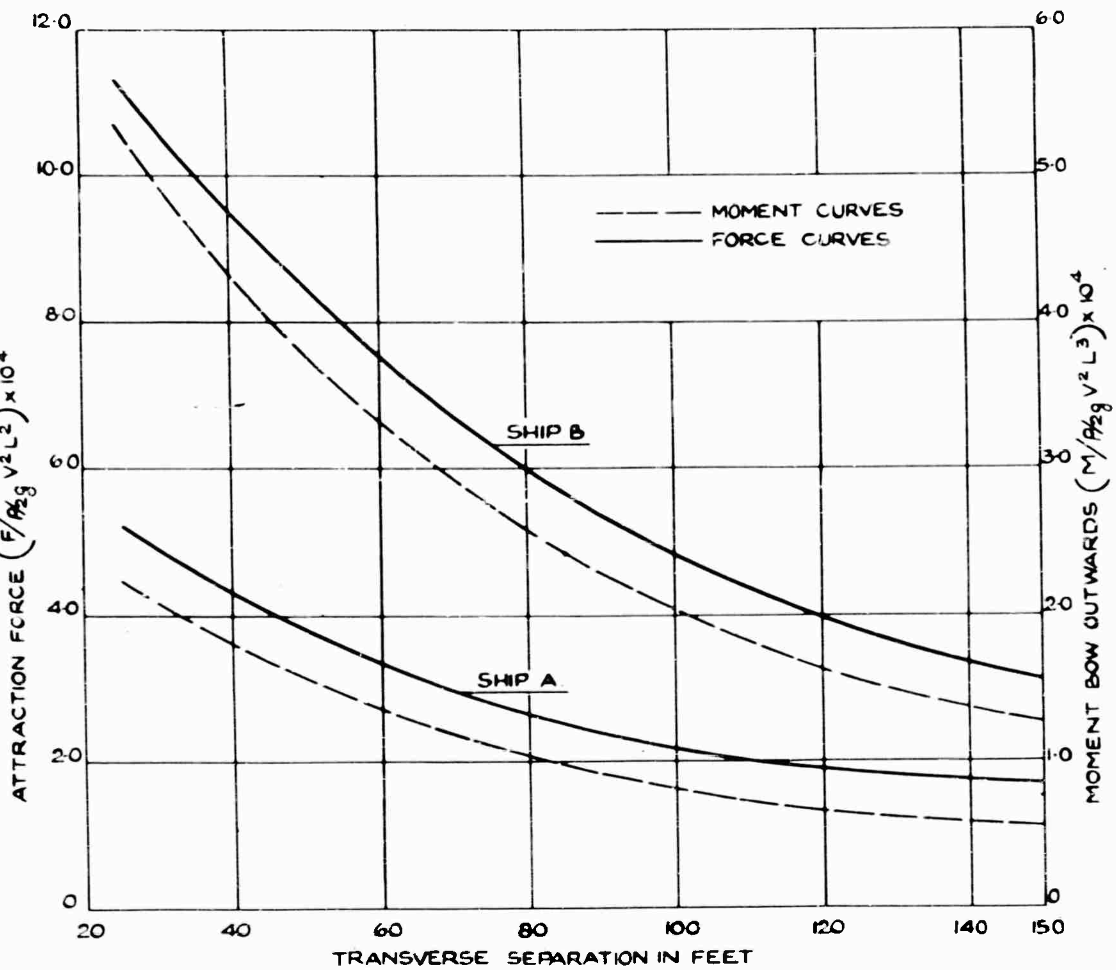

Figure 4 – Variation of Interaction Forces and Moments with Transverse Separation

Realization that these constrained model tests, whilst providing good indication of interaction during approach and breakaway, should be accepted with some reserve since the results apply strictly to ships at equal speed on parallel courses, led to the second series of tests with identical but freely propelled and remotely controlled models.

3. EXPERIMENTS WITH SELF-PROPELLED MODELS

The two models were controlled by the well-known "fishing line" technique which is illustrated diagrammatically in Figure 5 and needs no detailed explanation.

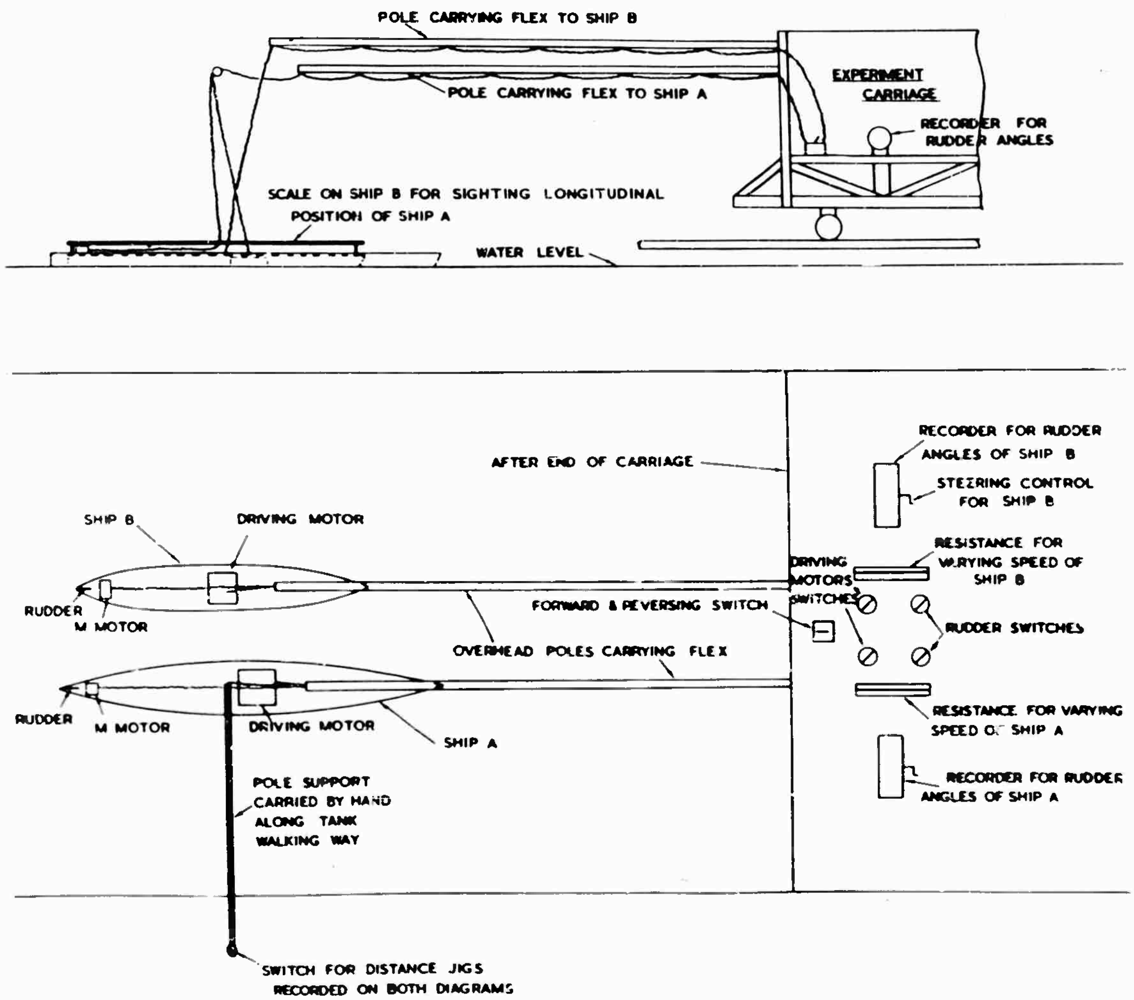

Figure 5 – Diagrammatic Arrangement of Self-Propelled Model Experiments

The models were complete with true-to-scale propellers and rudders. The propellers were driven by motors in the models, the speed of the motors being controlled by rheostats on the experiment carriage. The rudders were actuated by M-motors controlled from the carriage, and the controls were so arranged that the maximum speed of putting the rudder over

was approximately true to scale for the ships. The motion of the controls for the M-motors was recorded on drums on the experiment carriage, thus giving a continuous autographic record of the rudder angles used during the various maneuvers tested. The experiment carriage was run down the tank at various uniform speeds corresponding to from 10 to 20 knots, and Model B was maintained in constant relation to the carriage by appropriate application of the controls of the propelling motors and rudders. Course and speed of Model A were obtained by visual observation of the relative longitudinal position of the two models and the transverse position by sighting from the carriage. The longitudinal position was recorded on the same drum as the rudder records by an observer following the models down the walking way of the tank and closing a bell push switch as Model A passed various sighting marks on a graduated batten carried on Model B.

The following maneuvers were investigated for approach, "fueling," and breakaway, for transverse separations when fueling at 50 ft and 100 ft.

a. A overtaking B from stern to bow on a straight course.

b. A approaching B fine-on-the-quarter and taking up fueling position close aboard.

c. A breaking away from the close-aboard position on a divergent course and maintaining speed.

d. A on a parallel course well away from B, easing in to close aboard.

e. Runs with tho models maintained in various relative positions as in the constrained model experiments.

When in the fueling position there was no perceptible yaw in either ship, confirming deductions from the constrained experiments that the angle of yaw necessary in association with the rudder to balance the interaction forces and moments would be very small. Considerable movement of the rudder about its mean position was required, however, although the extreme angles were well within the maximum angle of 35 deg.

Considering the relatively short length of run available in the confines of the tank, and tho fact that the rudder operators had only 1/7 the time in which to anticipate and correct yaw compared with the helmsman in the full-scale ship (the models were made to 1/50 scale to minimize tank wall interference), the large fluctuations in rudder movement were not surprising.

In spite of these difficulties, however, the feasibility of the operations (a) to (d) was demonstrated conclusively and it was decided to carry out full-scale trials of the same type.

4. SEA TRIALS

The ships engaged in these trials were the replenishment ship BULAWAYO, battleship DUKE OF YORK, aircraft carrier ILLUSTRIOUS, cruiser SUPERB, and destroyer DUNKIRK. For the purpose of this paper, we are particularly interested in the interaction effects between BULAWAYO and DUKE OF YORK, but some observations pertaining to the other ships and relevant to the subject under review are included.

Each warship in turn was to approach the replenishment ship by the astern method (overtaking on a close parallel course), by the abeam method (overtaking on a parallel course far out until abeam and then closing in whilst maintaining station abeam), and fine on the quarter (overtaking from far out on a convergent course). After maintaining station abeam for about 20 minutes at speeds of 12, 15, 18, and 20 knots, the warship was to break away either by reducing speed and dropping astern on the same course or by turning away on a divergent course whilst maintaining or reducing speed.

In each ship the rudder angle was recorded autographically and the compass bearing, distance apart, rpm, and speed by log noted at regular intervals. The depth of water in the trials area averaged 30 fathoms, and the weather and sea conditions were generally moderate.

The trials led to the following general conclusions:

a. Approach from abeam is the safest method and has the advantage over approach from astern or fine-on-the-quarter that speed can be adjusted on the parallel course far out, so that, when the ships close, there is no tendency for the approaching ship to surge ahead or astern of the abeam position. The last two methods are, however, more expeditious and would be advantageous in small warships which are more maneuverable and can change speed more quickly than large ships. Approach fine-on-the-quarter is considered preferable to that from astern, in view of the rapid changes in the directions of the interaction forces with fore-and-aft position.

b. The best method of breaking away is by turning off on a divergent course, either maintaining or reducing speed. Breaking away by reducing speed and falling astern on a close parallel course would be practicable only in a smaller more maneuverable warship, but even in this case has no obvious advantage.

c. Approach and breakaway maneuvers and maintenance of the close-aboard positions can be carried out with equal facility between 12 and 20 knots in moderate weather conditions. In rougher weather, the trials with BULAWAYO and SUPERB indicated that a wide fluctuation in angles of rudder carried by the ships is to be expected. Thus, although the mean angles carried would be small, the amplitude of variation necessary to maintain course (particularly with wind and sea on the quarter) would prevent the use of the higher speeds in this range.

d. It was not possible reliably to assess the effect of the interaction on the speed of the ship, but only in the case of the destroyer was there any appreciable variation with change of separation, the destroyer tending to drop astern as the distance closed, and vice versa. The records of propeller revolutions, however, showed that the smaller of the ships engaged, in general, carries less revolutions than necessary for the nominal speed, suggesting that it is receiving assistance from the wake of the larger ship.

Comparison with the model experiments is possible only in the case of the DUKE OF YORK and BULAWAYO. The DUKE OF YORK was a sister ship of KING GEORGE V (Ship A) and BULAWAYO (580 ft by 72.5 ft by 25 ft by 20,000 tons) was of similar size and form to the OLNA (Ship B). The comparison is also limited to the correcting rudder required

by both ships in the abeam position, as the DUKE OF YORK used the abeam approach as opposed to the astern approach simulated by the model tests. A summary of the four runs carried out is given in Table 2.

TABLE 2

Run	1		2		3		4	
Mean separation, beam to beam, ft	125		110		105		115	
	A	B	A	B	A	B	A	B
Speed deduced from rpm, knots	12.3	12.0	15.5	15.0	18.2	18.0	19.7	20.0
Inboard rudder angle, deg Min.	0.0	9.0	0.0	7.5	0.0	4.5	0.0	1.0
Inboard rudder angle, deg Mean	5.0	13.0	6.0	12.0	6.5	10.0	6.0	8.0
Inboard rudder angle, deg Max.	13.5	17.5	16.0	16.5	15.5	14.0	15.0	14.5
Rudder angles deduced from model experiments, deg	4.5	8.0	5.0	9.0	5.5	9.0	5.0	8.5

The minimum and maximum rudder angles quoted for the ships are the average figures neglecting abnormal values. The rudder angles for the models were deduced from the constrained model experiments as described in Section 5. As the table shows, correlation between mean rudder angles estimated from the model experiments and those used by the ships is quite close. Typical records of rudder angle used by both ships when proceeding close aboard are given in Figure 6, and these bear out the wide variation in correcting rudder found necessary with the self-propelled models, although those on the ship were not so large or erratic, for the reasons previously stated. It will be appreciated that since these early sea trials were carried out, the technique of controlling ships close aboard has improved greatly and nowadays the variation in angle of rudder used is quite small.

It is interesting to record some other observations during these trials with DUKE OF YORK and BULAWAYO which are pertinent to the subject under discussion.

(i) Although the battleship approached the replenishment ship by the abeam method only, BULAWAYO reported a noticeable effect on steering, even at the lower speeds, as the bow of the DUKE OF YORK passed the stern of BULAWAYO several hundred feet away.

(ii) The propeller revolutions of BULAWAYO were quite steady during each run at the values appropriate to the nominal speed of the run. Those of DUKE OF YORK also were steady, but generally corresponded to slightly higher speeds than the nominal speed.

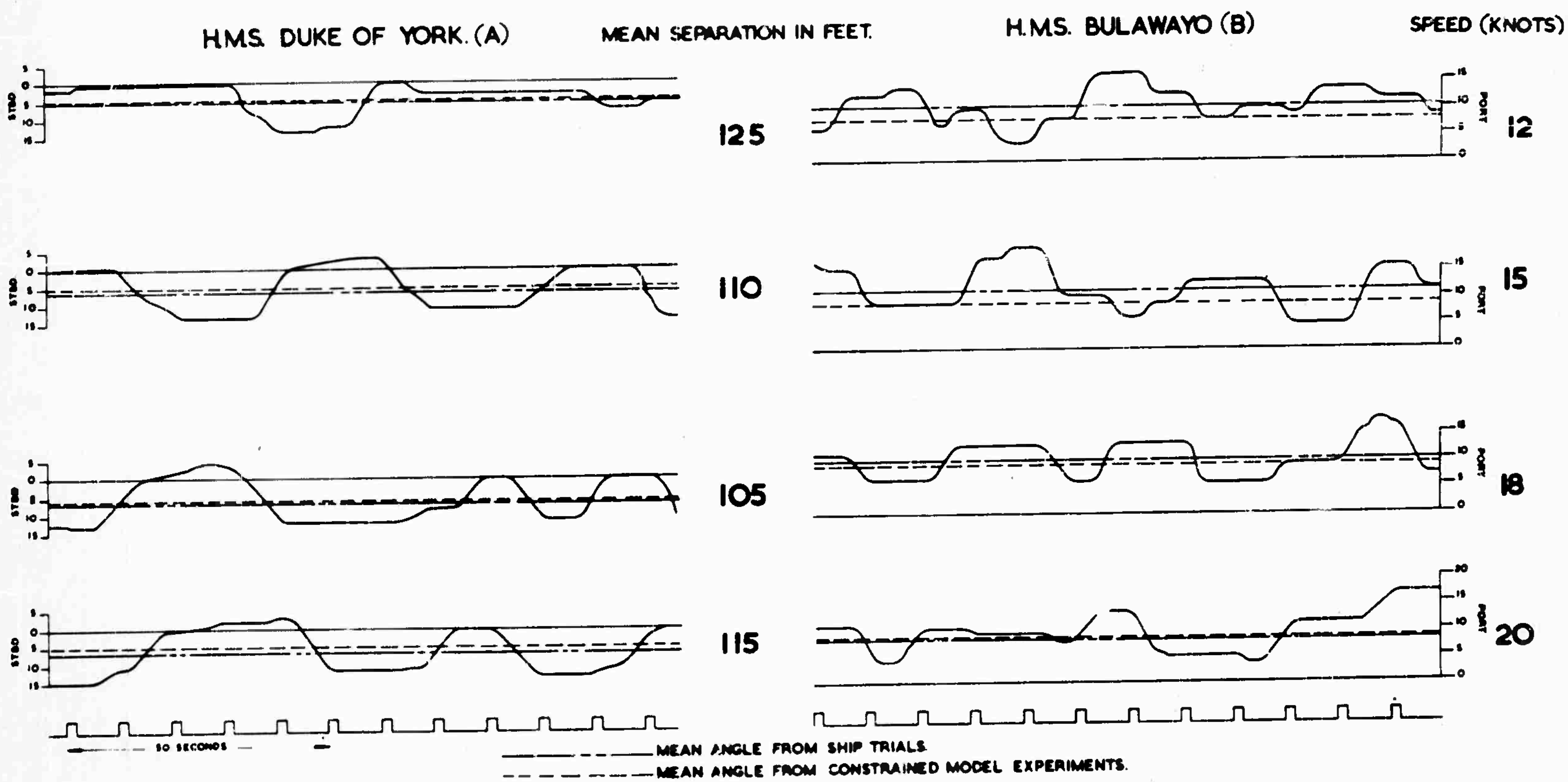

Figure 6 – Typical Rudder Records during Ship Trials; Close Aboard Position

(iii) The transverse distances between ships varied during each run, sometimes appreciably. The closest approach occurred during a 20-knot run when the distance closed to 55 ft. BULAWAYO then sheered off course away from DUKE OF YORK at about 4 deg, but within a minute the distance was increased to 120 ft and the course restored.

(iv) The ships maintained station with little fore-and-aft variation at all speeds.

(v) The mean angles of inboard rudder carried by BULAWAYO decreased as the speed increased, from 13 deg inboard at 12 knots to 8 deg inboard at 20 knots. The mean angles of rudder carried by DUKE OF YORK, however, were approximately constant, namely 6 deg inboard, at all speeds. On two occasions, the angles of rudder carried by BULAWAYO increased momentarily considerably beyond the average maximum values given in Table 1; viz., to 23 deg and 24 deg during the 12- and 20-knot runs when the ships closed to 60 ft and 55 ft, respectively, at these speeds. The rudder angle records of DUKE OF YORK do not show any corresponding abnormal increase at 12 knots, but at 20 knots the angle was increased momentarily to 21 deg.

(vi) Although the arrangements for measuring relative yaw between the ships in the abeam position were not very satisfactory, it was clear that each ship carried a bow-outward yaw of small magnitude, thus again bearing out the conclusion reached from the constrained model experiments.

5. INTERPRETATION OF DATA

In the first place, it is necessary to explain the method of presentation of the force and moment curves in Figures 2 and 3. As already noted, for each position of one model relative to the other, two forces on each were measured, at the positions of the guides; i.e., F_1 and F_2 in Figure 7. These two forces can be represented by a single force $F_l = F_1 + F_2$ and a couple M_l the magnitude of which depends upon the position about which moments are taken. It is convenient to choose a position for F_l conciding with this position and which is related to the action of the corrective rudder which must be applied to keep the ship on course and which is also associated with the force and couple set up as a result of any small yaw which the ship may take up. Such a point is the "neutral point" N at which, neglecting transient effects, any lateral force applied will not cause a change in heading although it will cause a change of course. The neutral point has been shown by experiments at AEW

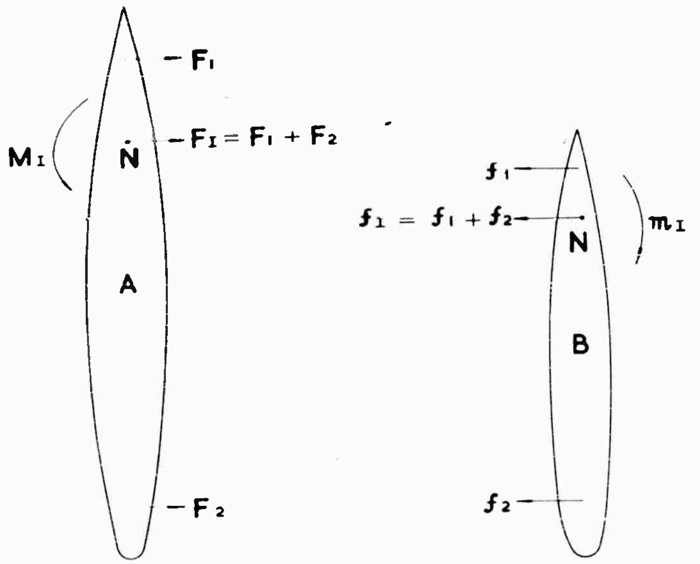

Figure 7 – Forces and Couples Due to Interaction

and from data given by Rydill[7] to lie well forward of the center of gravity, and has been taken at 1/5 the length of ship from the bow. The position of the neutral point is assumed constant for small angles of yaw but varies with the type of ship. It should be pointed out, however, that if a different position were chosen for this neutral point, say 0.25 to 0.15L from the bow, this would make no difference in principle to the discussion on use of corrective rudder which follows later.

Referring to Figure 7, it will be seen that if no corrective rudder action is applied the interaction moment will cause the ship to yaw outwards, bringing its stern towards the other ship. As the ship yaws, a hydrodynamic force due to it will come into action at the point N, as shown in Figure 8. If, now, correcting rudder is applied as shown, this will counteract the interaction moment and bring the ship back to a position of equilibrium at a small angle of yaw to the direction of advance. In this position, assuming a steady-state motion, we have for equilibrium:

$$F_R + F_Y = F_I \tag{1}$$

$$F_R \times \lambda L = M_I \tag{2}$$

where λL is the distance of the center of pressure of the rudder from the neutral point.

It will be appreciated that the above simplified analysis involves other assumptions than those already quoted, the more important being:

a. The system may be treated as one of steady motion.

b. The interaction forces and moments are unaffected by the action of the propellers.

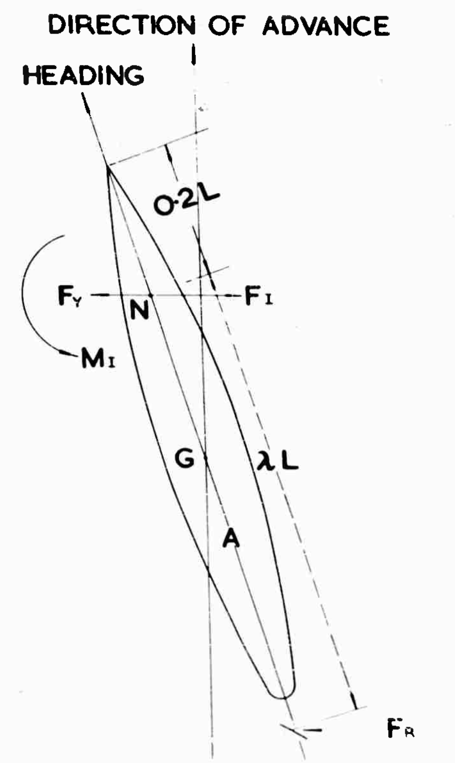

Figure 8 – Forces and Couple Due to Interaction; Rudder and Yaw

c. The interaction forces and moments are unaffected by the small drift angle taken up by each ship or by the movement of the rudder.

Assuming the validity of Equations [1] and [2], it is seen that, although the moment of the force from the rudder can counteract the interaction moment, there may be occasions when the rudder force will be insufficient to balance the interaction force unless the ship yaws slightly to produce a lateral force opposing the interaction force.

For moderate rudder angles the transverse force produced by the rudder can be expressed as

$$F_R = 0.03\ AV^2\theta$$

where A is the rudder area in square feet,

V is the ship speed in feet per second, and

θ is the rudder angle in degrees.

Substituting for F_R in Equation [2] gives the relation

$$\theta = \frac{M_I}{0.03\lambda AV^2 L}$$

and for Ship A $$\theta = 74{,}300 \left[\frac{M_I}{\rho/2 \; V^2 L^3} \right] \qquad [4]$$

Ship B $$\theta = 50{,}400 \left[\frac{M_I}{\rho/2 \; V^2 L^3} \right] \qquad [5]$$

Using the moment curves, the values of correcting rudder θ have been calculated from these formulas and are plotted in Figures 2 and 3. The direction of the correcting rudder required is shown in the small diagrammatic drawings in Figures 2 and 3.

It is then found, however, that the rudder force estimated from Equation [2] is not always large enough to balance the interaction force, and to effect a complete balance each ship must yaw slightly to produce an opposing interaction force. For instance, when the ships are abeam there is a force of attraction between them and couples tending to swing their bows apart (as Figure 7 and position 4 in Figures 2 and 3). The rudders will therefore be required to be turned inward, but the ships will settle down at a small outward angle of yaw (as Figure 8). The yaw angles for both ships have been estimated approximately, using the results of some recent experiments at AEW in which lateral forces and moments were measured at different speeds and angles of yaw on a mathematically shaped model. The results are tabulated in Table 3, for a speed of 10 knots at 100-ft separation.

TABLE 3

Item	Ship A	Ship B
Interaction Force F_I, tons	60	78
Rudder Angle θ, deg	5.5	9.8
Rudder Force F_R, tons	30	36
Yaw Force $F_Y = F_I - F_R$, tons	30	42
Yaw Angle (estimated), deg	1	1.5

The above figures are necessarily approximate, but they serve to indicate that, in spite of the large forces involved, both rudder angle and yaw angle are small, provided the ships are not allowed to approach one another too closely.

Referring to the small diagrammatic figures in Figures 2 and 3, it will be seen, however, that there are positions when both the interaction force and couple are tending to draw one ship into the other. Such positions are three for Ship A and five for Ship B. If now correcting rudder is applied to counterbalance the interaction moment, the rudder force adds further to the force of attraction. In these positions it is necessary to apply sufficient outboard rudder to overbalance the interaction moment and produce an outward yaw so that a yaw force is again introduced which will counteract both the interaction force and the rudder force. These conditions persist, of course, over an appreciable length of travel and are not merely momentary.

During this transient period, although the ships may be drawn closer together transversely initially if the rudder applied is sufficient to produce a large enough yaw force, then the ships should avoid collision provided always, of course, that the initial transverse separation is not so small that the available rudder cannot correct the inward swing caused by the interaction couple.

It will be seen that these positions three and five are quite close to the fully abeam position when opposite rudder has to be applied to keep the ships on course. In a short space of time, therefore, the rudder has to be swung from outboard to inboard, and the moment when this has to be done is obviously not easy to choose so that in these positions there is considerable navigational difficulty and risk.

If, when approaching positions three and five, the lateral separation of the ships is so small that there is insufficient rudder available to counteract the interaction moment or insufficient time for it to take effect, it would seem that a collision is unavoidable except that any reduction of speed on the part of the astern vessel, thereby reducing the interaction effects, must reduce the risk. In brief, the onus for avoiding action would appear to rest upon the astern ship, except that the ahead ship can probably ease the situation by breaking away on a divergent course.

In the absence of similar investigations into the case of ships passing in opposite directions, one can only conjecture as to the degree of risk. From first principles the same type of interaction occurs but the interaction forces act for a shorter time and are therefore less likely to bring about a collision.

The case of two ships which find themselves close and traveling in the same general direction on converging courses would likewise require separate and more complex investigation, and yet some guidance, if not comfort, can be derived from the experiments with self-propelled models and sea trials described previously. The safest action in such a case would seem to be for both ships to reduce speed as much as possible and lessen the convergence rather than that they should maintain course or that either should try to increase the convergence and cross the path of the other. This was clearly demonstrated by the model experiments described in Robb's paper.[3] Although such action would constitute a departure from the international regulations for navigation at sea, it is permissible under these regulations when collision appears probable.

These observations assume calm weather conditions. In inclement weather, as indicated by the sea trials carried out with a cruiser and replenishment ship, the risks involved would be increased. The larger variation in rudder angle alone must make control more difficult and the direction of wind and sea probably more so.

6. CONCLUDING REMARKS

As already intimated in the introduction to this paper, the discussion which followed Taylor's paper in 1909 introduced some fallacious arguments which have since been reiterated. For instance, one speaker, basing his remarks on experience on the Great Lakes, was convinced

that the propeller is the most potent of all the factors causing interaction effects. A few comments on this particular aspect, in the light of the results of the investigations under review, would not be out of place.

When a propeller is working, it modifies the pressure field around it and must therefore influence interaction effects to some extent. Measurements of pressure in the region of the propeller, however, show that the pressure changes due to it fall off rapidly with distance and would not, therefore, significantly affect interaction. It is notable that a propeller only 1 in. off the false bottom in the No. 2 Ship Tank at AEW gives the same thrust and torque as in deep water. As another indication of the smallness of propeller effect, the augment of resistance which is due to suction ahead of the propeller seldom exceeds about 15 percent of the resistance, so that the unbalanced component arising from this suction when modified by a ship in the vicinity would be expected to be almost negligibly small.

Doubts have also been cast as to the significance of interaction effects in deep water, as opposed to shallow water. For instance, in Reference 3, one contributor to the discussion stated that "the practical seaman regards interaction in deep water as an interesting legal argument which is not borne out in practical experience." This statement was made in spite of its being clearly demonstrated by the remote-controlled model experiments reported in that paper that interaction effects can have the most serious consequences. The experiments referred to, however, gave no indication of the general nature and magnitude of interaction forces and moments. It is felt that the results of the model and full-scale tests now reported should leave no doubt as to the significance in magnitude and varying nature of the interaction effects, nor as to their influence in cases of collision which have occurred and may occur again.

It is abundantly clear from the results that the magnitudes of the force and couple on each ship vary approximately as the speed squared, approximately inversely as beam-to-beam transverse separation, and oscillate in sense with longitudinal separation. These forces and moments begin to operate from before the time when the ships begin to pass until some time after they have passed. In the course of passing, the forces and moments undergo changes in magnitude and sense that can, generally speaking, be counteracted by rudder action, provided the ships are not so close that there is insufficient rudder angle available to do this, and that there is time for the rudder to take effect.

Broadly speaking, if two ships find themselves on close and more or less parallel courses for any reason, such as poor visibility, the sequence in corrective rudder angle required by both ships appears to be outward to inward to outward as one begins to overhaul, comes abreast of, and then overtakes the other, respectively.

Perhaps the most important aspect of the investigation is that during this process there are two positions of one ship relative to the other in which application of rudder to prevent her swinging across the course of the other adds to the force of attraction. In the case of the two ships considered in this report, one considerably larger and longer than the other, this occurs

in the longer ship when her amidships point is in the range 1/3 to 1/6 of her length astern of the amidships point of the shorter ship. Similarly, in the case of the shorter ship the condition arises when her amidships point is in the range 1/3 to 2/3 of her length astern of the amidships point of the longer ship. The situation is, in fact, more serious for the smaller ship, as would be expected.

Whilst the results reported cannot be applied to the general case of ships of widely differing size and form, nor to the effect of inclement sea conditions, it can be claimed that they indicate that model experiments can at least establish guiding principles in this complex subject.

The new and larger facilities now becoming available, with their equipment for wave generation and modern techniques, present an opportunity of widening the scope of such investigations into interaction effects which should not be missed. The subject can be regarded as a special branch of the general subject of maneuverability, and since it is of international interest we may perhaps look forward to its being included, at some future date, in the deliberations of the Maneuvering Committee which is being set up by the International Towing Tank Conference in September this year.

ACKNOWLEDGMENT

The author is indebted to the Director General Ships, Mr. A.J. Sims, O.B.E., R.C.N.C., for permission to use the relevant data in this paper, gratefully acknowledges the assistance of those members of the staff at Haslar who have been concerned with its preparation, and wishes to express his pleasure at being invited to present it to this Symposium.

The paper is published with the approval of the Lords Commissioners of the Admiralty, but the responsibility for any statements of fact or opinion rests solely with the author.

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DISCUSSIONS

C. G. Moody:

Mr. Newton's paper is particularly welcome at the Taylor Model Basin where studies on the subject of interaction effects between ships during replenishment-at-sea operations are currently being conducted.

The technique employed in the Taylor Model Basin tests is essentially the same as that described in the paper—the principal difference being that at Haslar one model is spotted in successive positions along its course while the forces and moments are measured, whereas at the Taylor Model Basin the usual procedure is to run one model slowly past the other while the forces and moments acting on the models are recorded continuously. An advantage of the latter method is that some irregularities in the force-and-moment curves are brought out which might otherwise escape notice. These irregularities are largely due to wave effects. They are negligible in some instances but very pronounced in others.

Where the wave trains of the two vessels are superimposed along the side of one hull, there are appreciable pressure effects, which in some instances are augmented by the reflection of the bow wave of one vessel from the side of the other. It may be of interest to note in this connection that at high speeds, long wave hollows may be superimposed upon the reduced pressure, or "suction," field of one vessel at a considerable distance away from the other. Under such conditions the wave crests are likely to be especially steep and sharp, so that a small vessel may even broach-to in them and sheer in toward the other ship.

In the Taylor Model Basin tests, the relative motion of the models can be stopped or reversed, and the effect of different relative speeds can be investigated. The relative speed of approach or departure is a consideration of particular interest in the replenishment of small, fast, full-bodied vessels.

The Taylor Model Basin replenishment-at-sea tests have generally been run with the models self-propelled to represent the effect of the propeller slipstream on the rudder. The idea persists that the propeller has a significant effect upon the interaction of ships, and was brought up recently in connection with the operation of ships in the St. Lawrence Waterway.* There is some truth in it in the case of a ship in confined water. When a large bulk cargo vessel is proceeding under its own momentum along one bank of a canal with the rudder amidship, the effect of starting the propeller is to increase the velocity of the flow around the stern. On the side of the hull adjacent to the wall where the water is confined in a small space, the cross-section area of the flow is restricted and the velocity of the flow is increased. Vessels of this type generally have very full water lines aft, which deflect this flow away from the wall and thus produce a steering effect. In conjunction with the Venturi effect, this produces hydrodynamic force that tends to draw the stern in toward the bank. Nevertheless, where a rudder in the propeller slipstream is used to correct this tendency, the net effect of the propeller action is usually beneficial.

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*Hauck, P.F. and Connell, T.P., "Destroyer Seamanship in the St. Lawrence Seaway," The U.S. Naval Institute Proceedings, Vol. 86, No. 3, pp. 131–137 (Mar 1960).

A. Taplin:

Mr. Newton's explanation was extremely clear. However, I would like to ask a question as to what effect the sea has on the ability to handle ships during replenishment. Mr. Newton mentioned that the interaction force increases with the square of the speed. Offhand, it would seem that the force on the rudder and the force on the yawed hull would also increase with speed squared. On that basis alone we would have a little trouble, but there are two other effects of speed that come to mind. For example, when two ships are 100 feet apart and are going fast, they will close that distance more rapidly than they would have at slow speed. Also, the bow waves are extremely different at high speed. Specifically, is there anything other than the bow-wave effect and that the distance closes more rapidly, that makes high-speed replenishment difficult?

B. Silverstein:

I enjoyed seeing that the experimental data in Mr. Newton's paper exhibited the same trends as those shown in my paper on linear theory of ship interaction; that the lateral force diminished inversely as the distance between the ships. It also came out theoretically that the force would vary in an oscillatory manner with the longitudinal distance, and this was checked experimentally. I would be very much interested in trying to check in detail the results of the theory against the forces and moments you measured.

My paper ended with a rather complicated equation which did not lend itself to easy calculation. I appreciate the fact that I now have experimental data to confirm the equation.

You mentioned earlier that one of the key problems was not just calm water, but what is the behavior in a seaway? This is a very complicated addition to the theory but I think it is the important problem. We are beginning to understand about calm water, but the replenishment-at-sea operation in rough water is the difficult problem.

J. P. Breslin:

I was somewhat interested in the question that people have raised in the past about the possibility that the propeller might play somewhat of a role in the interaction effect. I would like to add a small voice to the opinion that Mr. Newton has ably expressed that certainly in deep water one should not expect the propeller to influence interaction very much since the pressure field decays so rapidly. I think a quick way of getting at this is to realize that the propeller thrust force is roughly equal and opposite to the drag of the ship, and the drag of the ship is a small fraction of the displacement, in the order of 2 percent, perhaps possibly in the order of 10 percent, at the most.

I would like to hear some mention as to how big these forces are; what their limits are (if the ships were allowed to touch) in fractions of the displacement; what order of magnitude of forces we are talking about.

H. E. Saunders:

As the result of a study having to do with the modernization of the Panama Canal which was carried out at about the same time that Haslar was doing the work described by Mr. Newton, the U.S. Army Engineers established the following requirement for a modernized canal: When one ship passes another, or performs any type of operational maneuver that could be assumed to happen in the course of operating in the canal, it should be possible for each ship to perform its share of the maneuver, barring accidents or any unforeseen circumstances, with not more than one-third of the total available rudder angle. In other words, if the total available rudder angle was 30 degrees, then the model tests should show (and the model tests eventually did show) that not more than 10 degrees of rudder would be required by the pilot (through the steersman and the controls) to handle the ship in the normal manner; leaving the other two-thirds of the rudder angle for emergencies and times when the commanding officer has to exercise his own discretion. Incidentally, as the ships get too close, there isn't anything the commanding officer or anybody else can do.

Although it is an operational feature, and perhaps has no part here, it is just as well to remember that in any operation of this kind, we must not count on using too much of the available rudder angle for normal operation. We have to leave plenty of reserve for the forces that we don't anticipate, or for the forces that become too large, or for some other effects such as wind or third ships and so on, which come into the picture.

Author's Closure

I was very pleased to hear what Mr. Moody said about the latest techniques at TMB. One should really make one model pass the other in order to get the transient effects, which obviously affect the situation, whereas in the Haslar experiments they were in definite positions.

The main point of what Mr. Moody said has to do with wave action. He developed this point by describing how the two waves of the ship marry, and said that you can get in dangerous conditions where the forces are quite large. This was borne out by the sea trials of the KING GEORGE V and the OLNA, and the figure I remember is that when the battleship approached the replenishment ship, 800 feet away, as her bow began to overtake the stern of the OLNA, the OLNA was forced to use 5 degrees of rudder to keep on the course. That's a long distance, and it was obviously a large force.

Mr. Taplin asked a rather awkward question: Is there any other factor other than the bow wave or the pressure field effect which can make this operation of replenishment more difficult? I would immediately say that, at high speed, you of course have far less time to take avoiding action if you start the maneuver wrong. But from the sea trials of the KING GEORGE V, it is perfectly clear that this operation can be done safely with some ships up to 20 knots. The sea trials and the model tests were done at 10 to 20 knots, and the same rudder angle, roughly speaking, was measured on both ships, whatever the speed within that range.

Dr. Silverstein confirms that the experiment was borne out in theory. I might say that not enough people realize that more comes out of the marriage of theory and practice than from out of either, and this is a beautiful example of that.

Dr. Breslin wants to know the magnitude of the forces when the separation between the ships is nought [zero]. The curves of Figure 4 of the paper are going up pretty well and are obviously asymptotic and theoretically reach an infinite force (I think that answers Dr. Breslin); it is a pretty big force.

Captain Saunders speaks from a wealth of experience. I did not know of the rule that one should keep two-thirds of the rudder available, especially in a canal. This was borne out by the description of an accident given by Prohaska in a discussion of Robb's paper.* The two ships were on parallel courses in passing; one had to veer away with full rudder and got clear—but it also rammed a pier; which makes your point admirably. One must have some capacity left in the rudder.

*Prohaska, C.W., Comments on "Interaction between Ships," by A.M. Robb, Transactions of the INA, Vol. 91 (1949), p. 337.

MODEL TESTS WITH BOW-JET (BOW-STEERING) SCREW PROPELLERS

by

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ABSTRACT

The paper deals with the steering effect from bow-jet screw propellers. This type of propeller is placed in a transverse tunnel in the bow and is used for maneuvering at low ship speed. Model test results are given for a controllable-pitch propeller with 4 planar blades at adjusted pitch ratios P/D = 0.4, 0.5, 0.7, and 0.9; and for a CPP with 3 planar blades at adjusted pitch ratio $P/D = 0.7$. Model test results are also given for a 4-bladed fixed blade propeller, pitch ratio $P/D = 0.7$. The model tests are compared with full-scale tests of two 300-HP controllable-pitch bow-jet propellers.

There is a note on forces and turning moment which occur on a normal screw propeller as a result of nonparallel flow and which affect the steering characteristics of a ship especially at low speed.

INTRODUCTION

Bow-jet propellers or bow-steering propellers, as they are also called, have been more widely used during recent years. Especially, they have been used on railway and car ferries and on buoy tenders; i.e., ships where maneuvering at low speed is often required.

A number of different types of bow-jet propellers have been designed. One of the first was the cycloidal propeller which, in the first installation, was mounted with the blades working freely under the ship's bottom. Later this propeller was placed in an athwartship tunnel, like other types that are mentioned below. The cycloidal propeller has been installed on, for example, the PRINCESS OF VANCOUVER, and the TRELLEBORG.

Other types are those operating with fixed-pitch screw propellers. The Pleuger system, for instance, consists of a fixed-pitch propeller in a circular duct operated by a submerged electric motor. The Jastram system also utilizes fixed-blade propellers but has two contrarotating propellers, one at each end of a propulsion pod to which power is supplied over a bevel gear. The former system is, for example, applied on the car ferry COMPIEGNE and the latter system on the buoy-tender WALTER KÖRTE.

The Gutche system essentially consists of two transverse ducts and a vertical fixed-pitch propeller. Magnitude and direction of thrust are controlled by a cylindrical valve driven by a separate motor. This system is used on the East-German railway ferry SASSNITZ.

Controllable-pitch propellers have also been used. The KaMeWa system consists of a controllable-pitch propeller placed in a circular duct and housed in a propulsion pod. Among others, the railway ferry PRINSESSE BENEDIKTE and the car ferry PRINSESSAN CHRISTINA utilize the KaMeWa system.

This paper deals with model testing of screw propellers used as bow-steering propellers, with the emphasis on controllable-pitch propellers. The general arrangement of a bow-steering controllable-pitch propeller is shown in Figure 1.

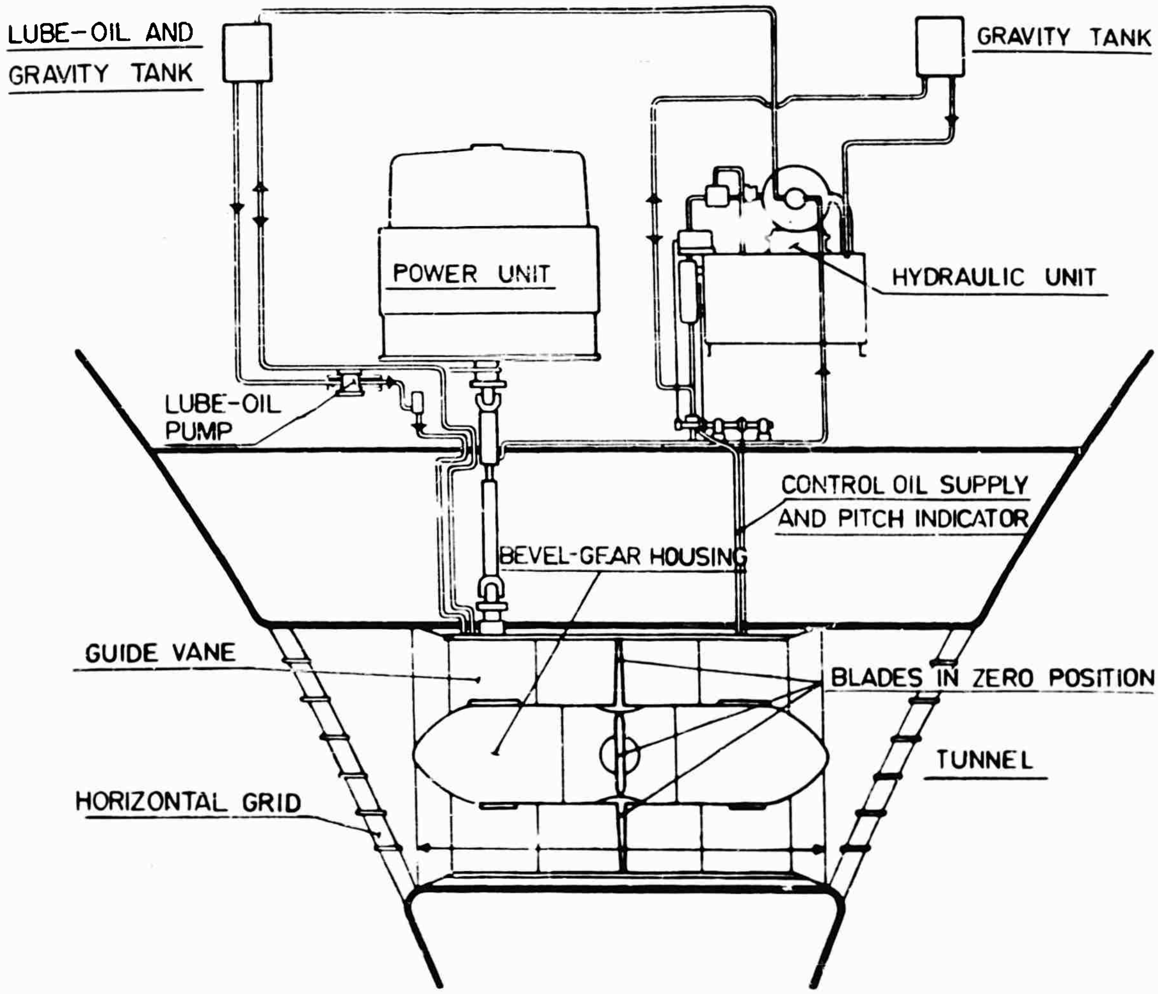


Figure 1 – General Arrangement of Bow-Steering Propeller

The propulsion pod is supplied with power from a vertical shaft coming in at one end of the pod where the bevel gear is located. At the other end, hydraulic oil is supplied to the blade-turning servomotor in the propeller hub. At this end also the feedback of the control mechanism is placed. It is noted from the drawing that the propeller has an unusually large diameter of the hub body. This is determined by the space required of the bevel gear.

There are three supports at each end of the pod. Only one at each end accommodates part of the mechanism.

MODEL TESTING ARRANGEMENT

The model testing of this bow-steering system was made at the Kristinehamn Cavitation Tunnel. In the tunnel, which has a measuring section of roughly 800 × 800 mm, a model of the duct around the propeller was built. At each end this duct was surrounded by a disk simulating the plating of the ship. The duct was connected to a scale so that axial forces on the duct could be measured directly. The arrangement is shown in Figure 2. Thrust and torque for the propeller were measured with the ordinary equipment of the model propeller shaft.

Data of model propellers used were:

Model Number	317-B	327-B	331-S	339-C
Diameter D, m	0.225	0.225	0.225	0.225
Number of blades	4	4	3	4
P/D	0 (planar) (blades)	0	0	0.700
A_E/A_O	0.500	0.432	0.450	0.432
Hub diameter d, m	0.0847	0.0847	0.0847	0.0847
d/D	0.376	0.376	0.376	0.376

The four models are of the Kaplan type; i.e., the blade tip has the form of a circular arc. The clearance between the blade tip and the tunnel wall is 1.5 mm. When the blade is adjusted for positive or negative pitch, this clearance, however, increases at the blade edges. Figure 3 shows the blade designs.

PRELIMINARY TESTS

The preliminary testing was made with Model 317-B. In one part of the tests the influence of various types of guide vanes was observed. The arrangement was tested with one, three, and six guide vanes at each end of the pod. One test was made with the guide vanes set symmetrically at 0, 5, and 10 degrees. In this case the arrangement with three guide vanes was used. The total thrust produced at a given horsepower was within 2-3 percent for all arrangements except for large angles where the thrust dropped. Therefore, it was decided to use the arrangement with three parallel guide vanes at each end of the pod.

The axial flow just outside the tunnel at a distance of about ¾ propeller diameter measured on the centerline of the tunnel was observed. Model 317-B, adjusted pitch ratio 0.70, was used. The velocities are plotted in Figure 4. It is noted that the velocities at the lower part of the tunnel are lower than in the upper part of the tunnel, to some extent depending upon the fact that the pitot tube has its longest distance from the tunnel end at the lowest part of the tunnel.

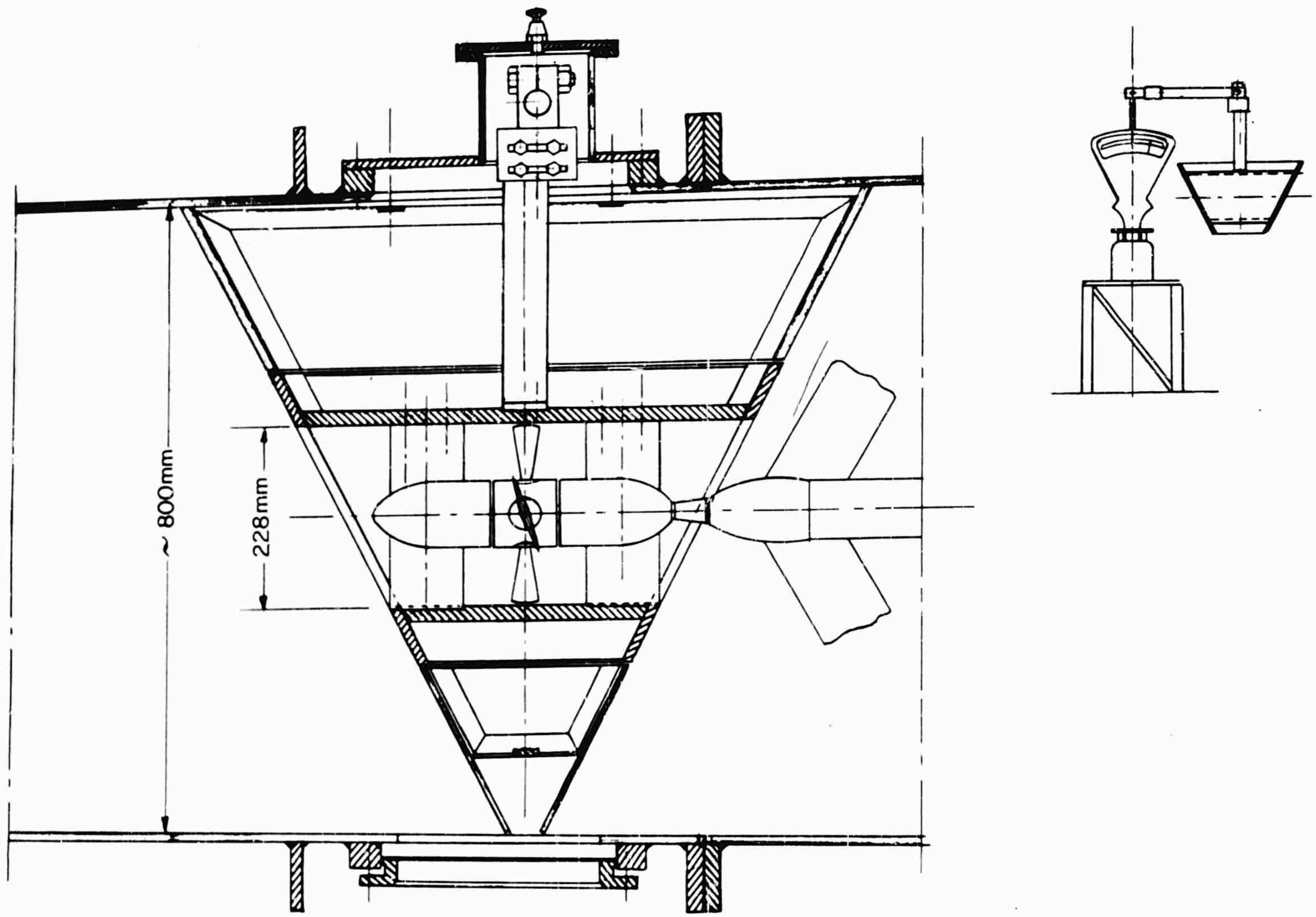
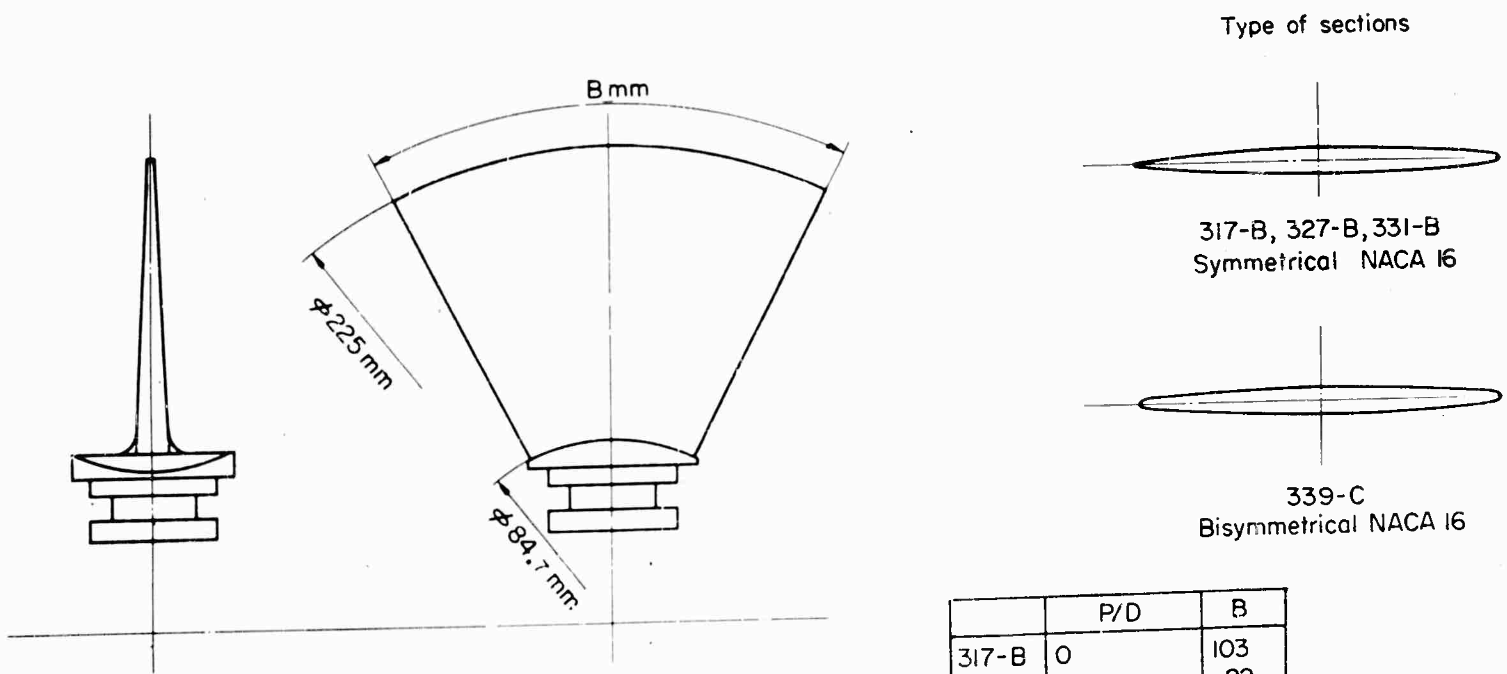


Figure 2 – Testing Arrangement



	P/D	B
317-B	0	103
327-B	0	82
331-B	0	123.5
339-C	0.700 const	82

Figure 3 – Model Propellers

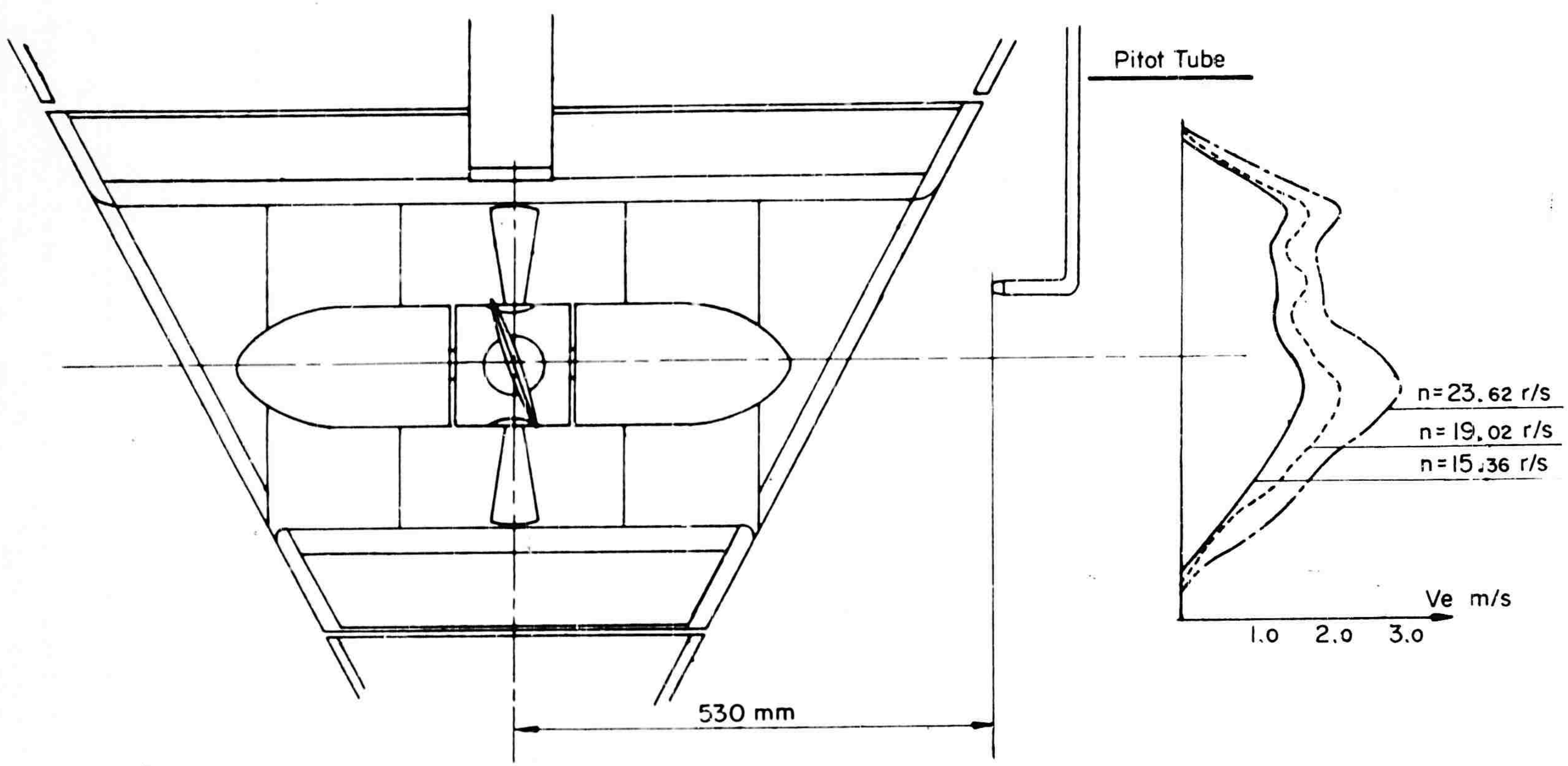


Figure 4 – Velocity Distribution at Tunnel Outlet

Streamline tests were also made and showed that the flow in the lower part of the tunnel was somewhat unstable. In order to improve this situation, a bulge was arranged at each end of the tunnel (Figure 5). This increased both K_T and K_Q for the propeller, but instead of being positive, the axial force on the tunnel turned negative.

A numerical example shows for Model Propeller 317-B converted to full scale, 1.3 m diameter and 285 DHP metric:

Test Number	15 (without bulge)	16/17 (with bulge)
P/D	0.7	0.59
RPM	389	389
T Propeller	kp 2816	3560
T Tunnel	kp + 409	− 840
Total Thrust	kp 3225	2720

This result is somewhat surprising. Further tests on the effect of bulges are planned.

Bulge and grid were tested separately. The grid has a stabilizing effect on the flow. Grid without edge ring was finally adopted.

CAVITATION INDEX

The cavitation index which has been used is:

$$\sigma^1 = \frac{p_0 - e}{\frac{\rho}{2} D^2 n^2};$$

Figure 6 gives this index for different combinations of diameter and shaft speed. The submergence of the shaft is assumed equal to the diameter.

MODEL RESULTS

The final tests were made with three and four guide vanes for 4-bladed and 3-bladed models, respectively. The openings of the tunnel were free; i.e., no grids were applied. Check tests show that if a grid is used, the total thrust increases by 1.5–2.0 percent. Figures 7–10 show the test results.

The flow set up in the cavitation tunnel by the model propeller was stopped by turning the tunnel impeller slowly in reversed direction. The total thrust is produced by the propeller and partly also as lift on the guide vanes. For the sake of simplicity, these forces are added in one K_T total value.

It is noted that around 10 percent of the total thrust is contributed by the guide vanes.

(Text continued on page 39.)

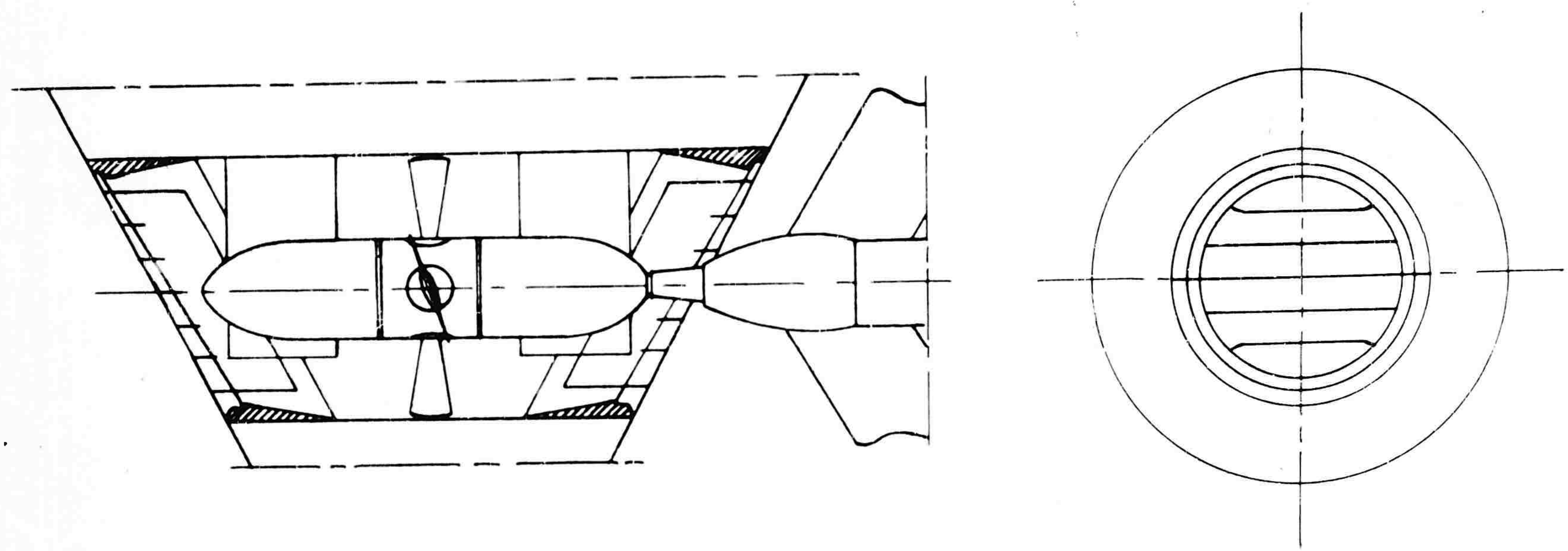


Figure 5 – Arrangement of Bulge and Grid

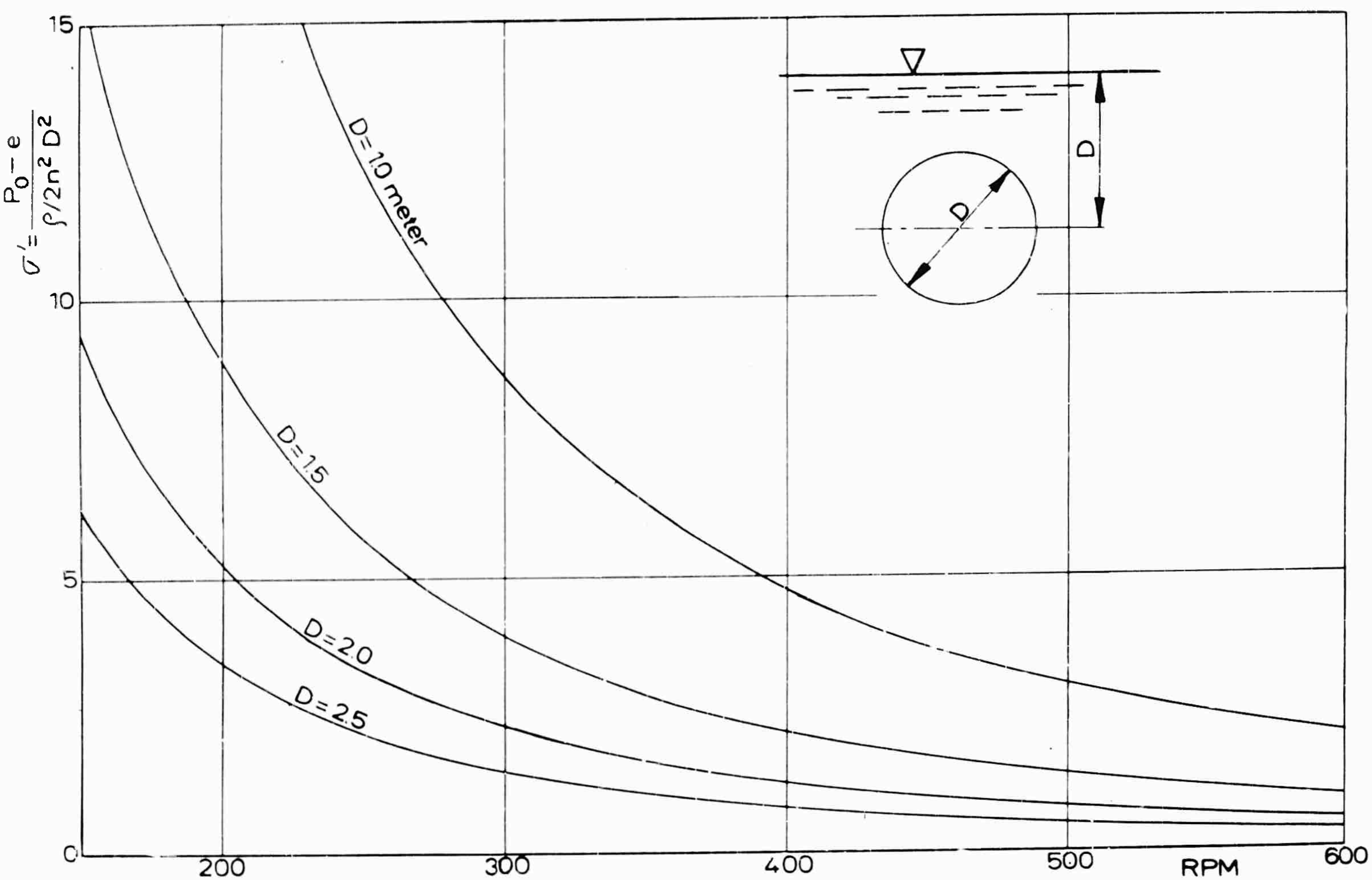


Figure 6 – Cavitation Index

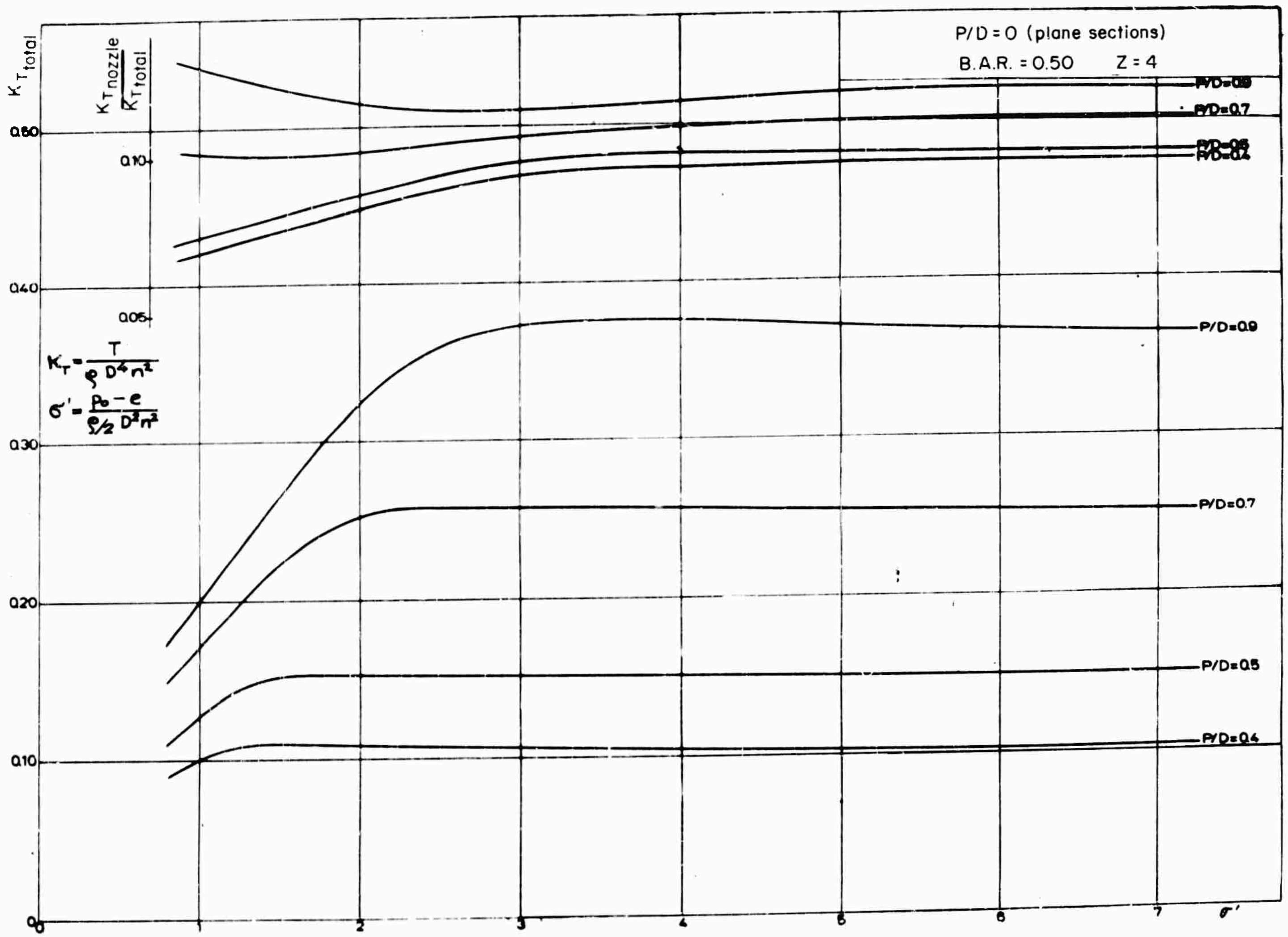


Figure 7 – Model 317-B

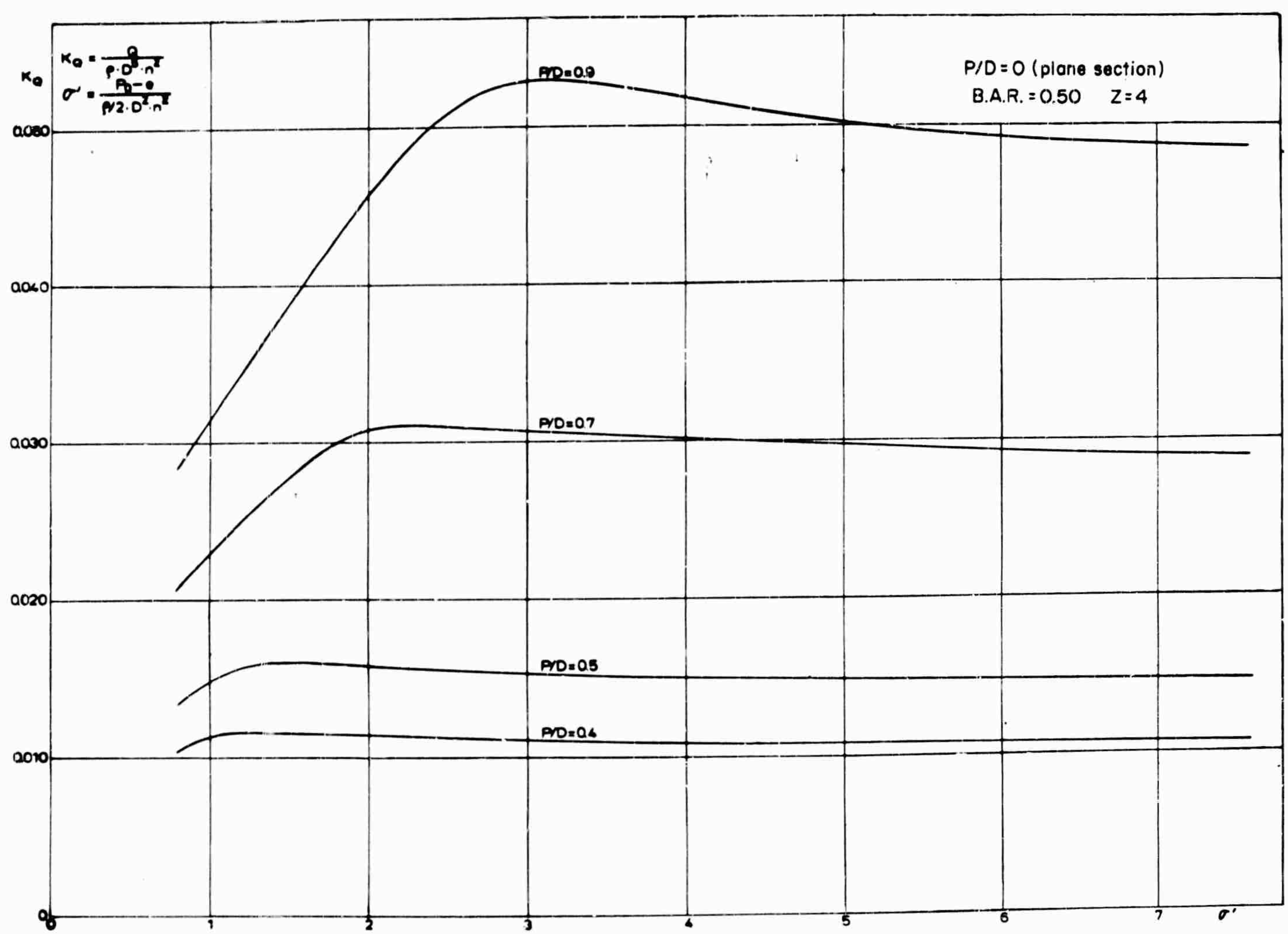


Figure 8 – Model 317-B

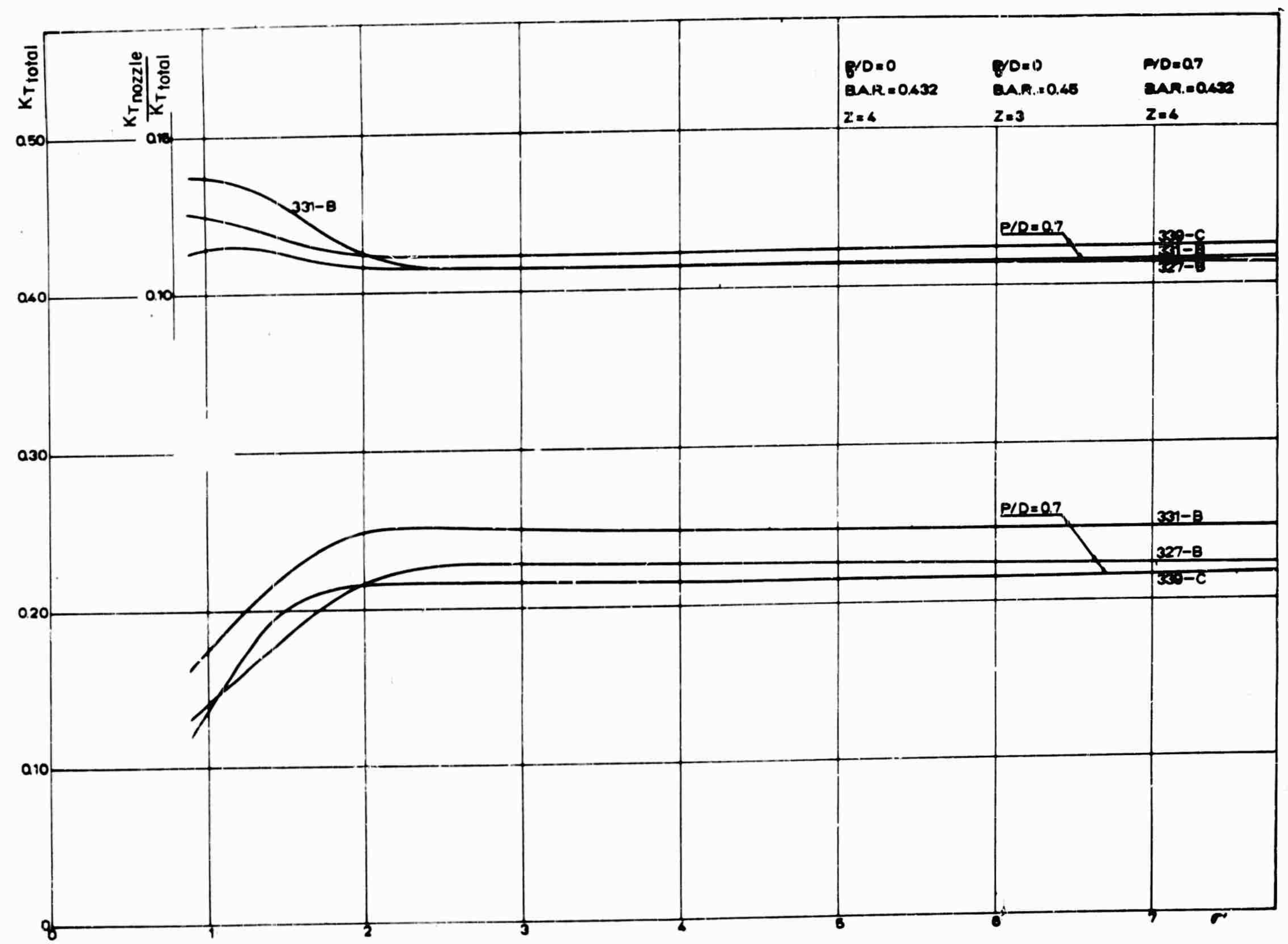


Figure 9 – Models 327-B, 331-B, 339-C

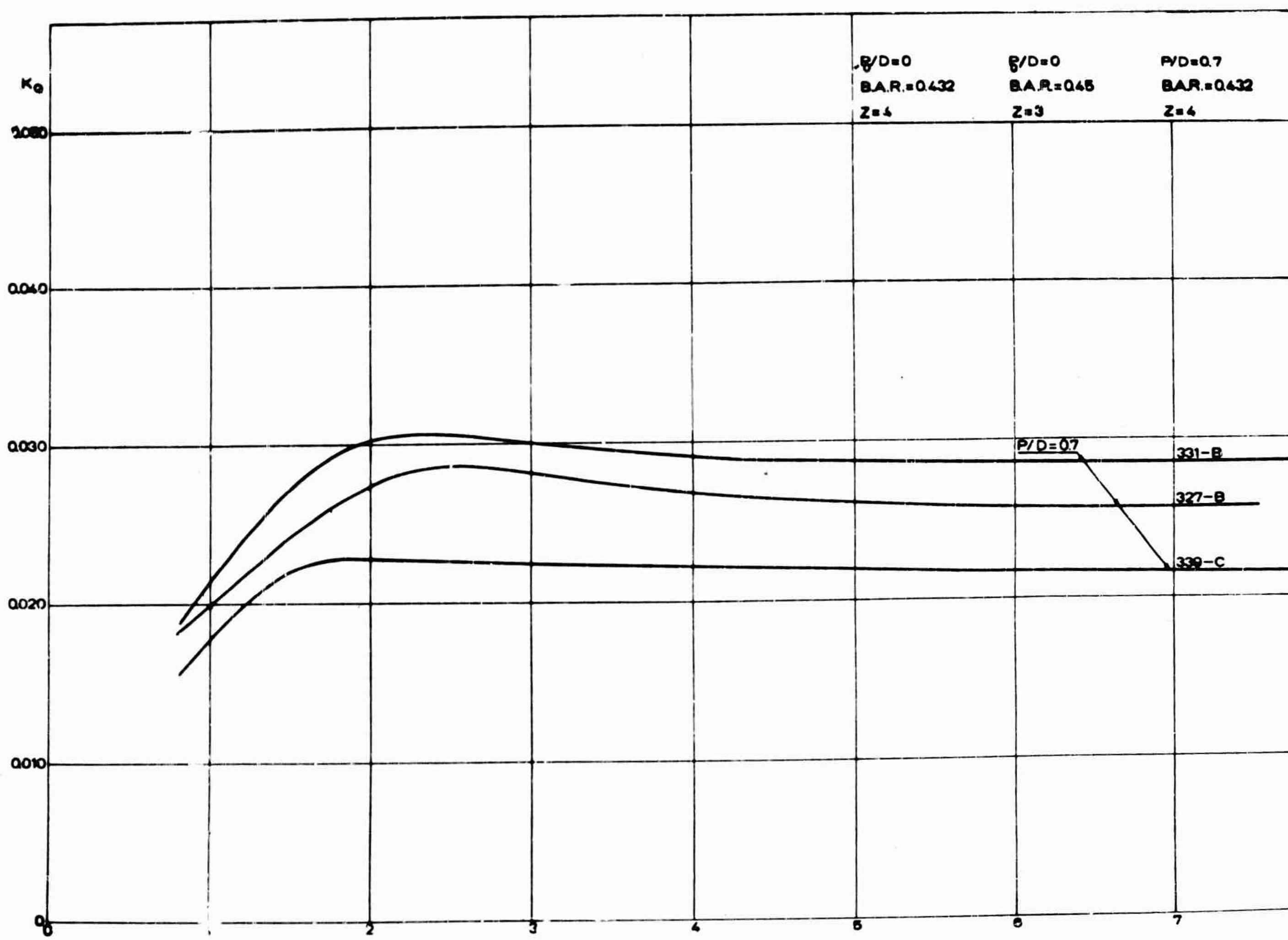


Figure 10 – Models 327-B, 331-B, 339-C

A comparison of the four models is made (Figure 11). This shows: The 3-bladed 45-percent BAR Model 331-B is just slightly less efficient than the 4-bladed 50-percent BAR reference Propeller 317-B. With respect to cavitation, there was no significant difference.

The 4-bladed Model 327-B with BAR 43.2 percent is considerably inferior to 317-B, as K_T total is reduced by 6.5 percent. Model 327-B differs from 317-B only with respect to BAR. As could be expected, 327-B showed more cavitation.

The fixed-blade propeller 339-C, which differs from 317-B mainly in having lens-shaped sections and a normal helicoidal surface, is better than 317-B. The K_T total-value is 3.9 percent larger. Considering that the CP-propellers have planar blades and thus an unfavorably high load at the tips, this superiority of the fixed-blade propeller may seem small. However, as in a Kort nozzle, the tunnel walls reduce the loss over the blade tips, and this effect is more pronounced for the propeller with highly loaded blade tips.

The cavitation picture was somewhat better for 339-C than for the other propellers.

There was some difficulty in taking photographs showing the cavitation clearly. Therefore, small drawings were also made. Figures 12 and 13 show the cavitation pattern.

It is noted that as usual at static pull the cavitation appears as sheet cavitation extending toward the boss from the blade tips. The guide vanes were cavitation-free.

INCREASE OF SHIP RESISTANCE DUE TO BOW TUNNEL

A certain increase in resistance must be accepted using a bow propeller Two examples are given:

1. Passenger ship, L_{pp} = 82.00 m, block coefficient = 0.575, draft = 4.1 m, d_{tunnel}/draft = 0.32

V knots	16	16.5	17	17.5
Increase in DHP percent	0.9	1.6	1.7	1.7

2. Bulk carrier, L_{pp} = 217.0 m, block coefficient = 0.86, draft = 8.1 m, d_{tunnel}/draft = 0.23

V knots	11	12	13	14	15	16
Increase in DHP percent	0	0.2	0.4	0.4	0.5	0.8

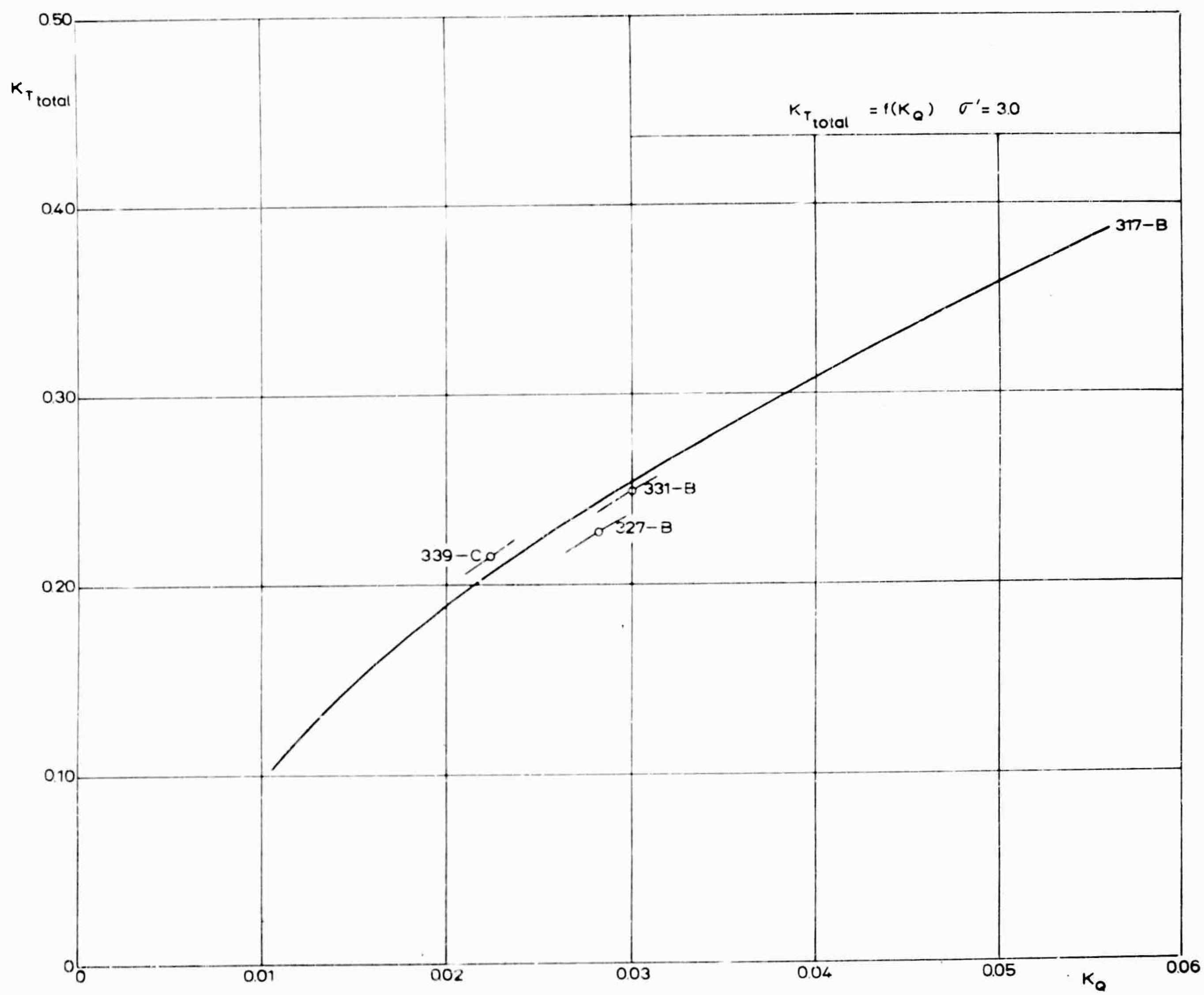


Figure 11 – Model Tests With KaMeWa Bow Steering Propellers

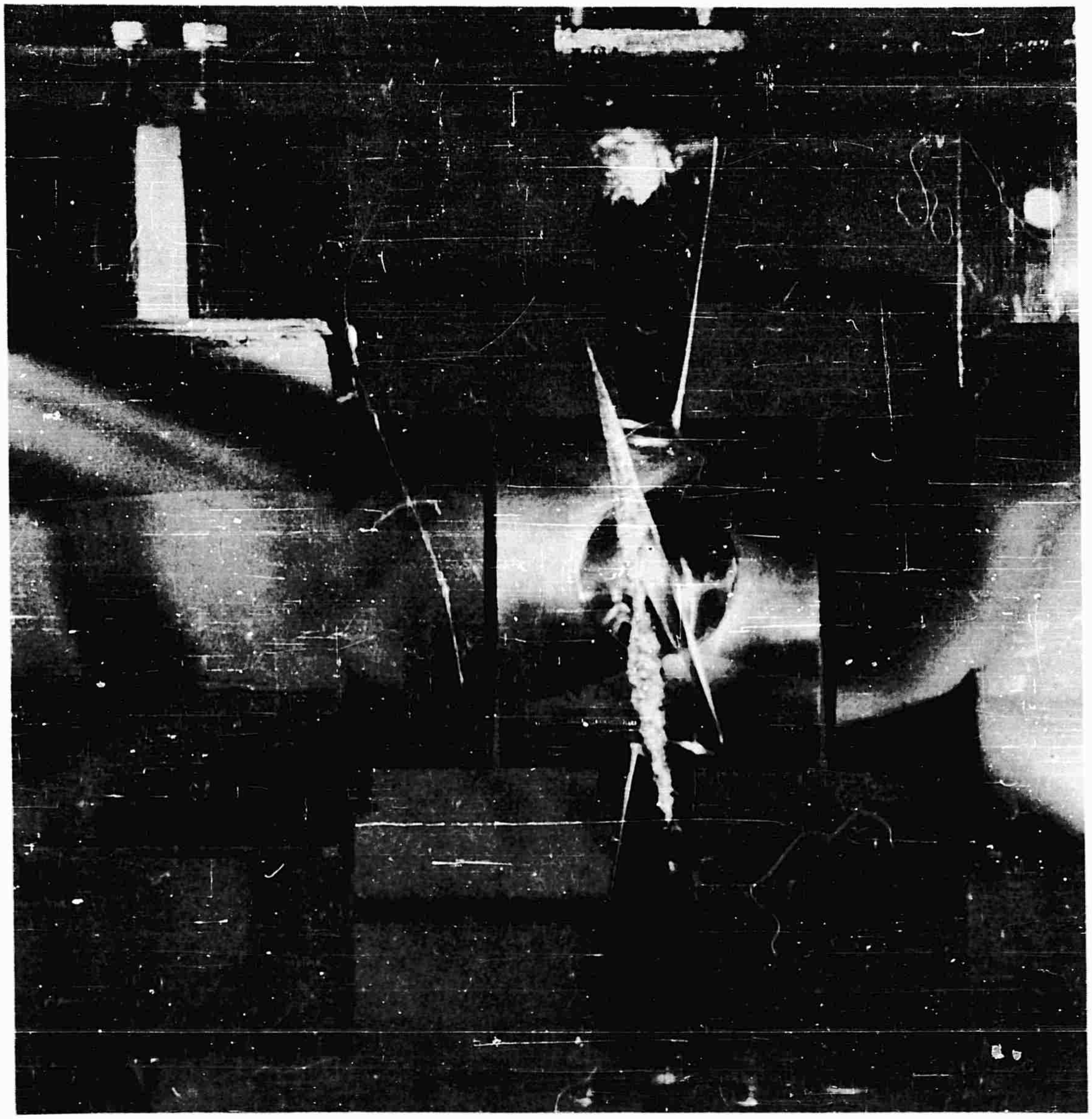


Figure 12 – Model 317-B, $P/D = 0.700$, $\sigma^1 = 2.78$

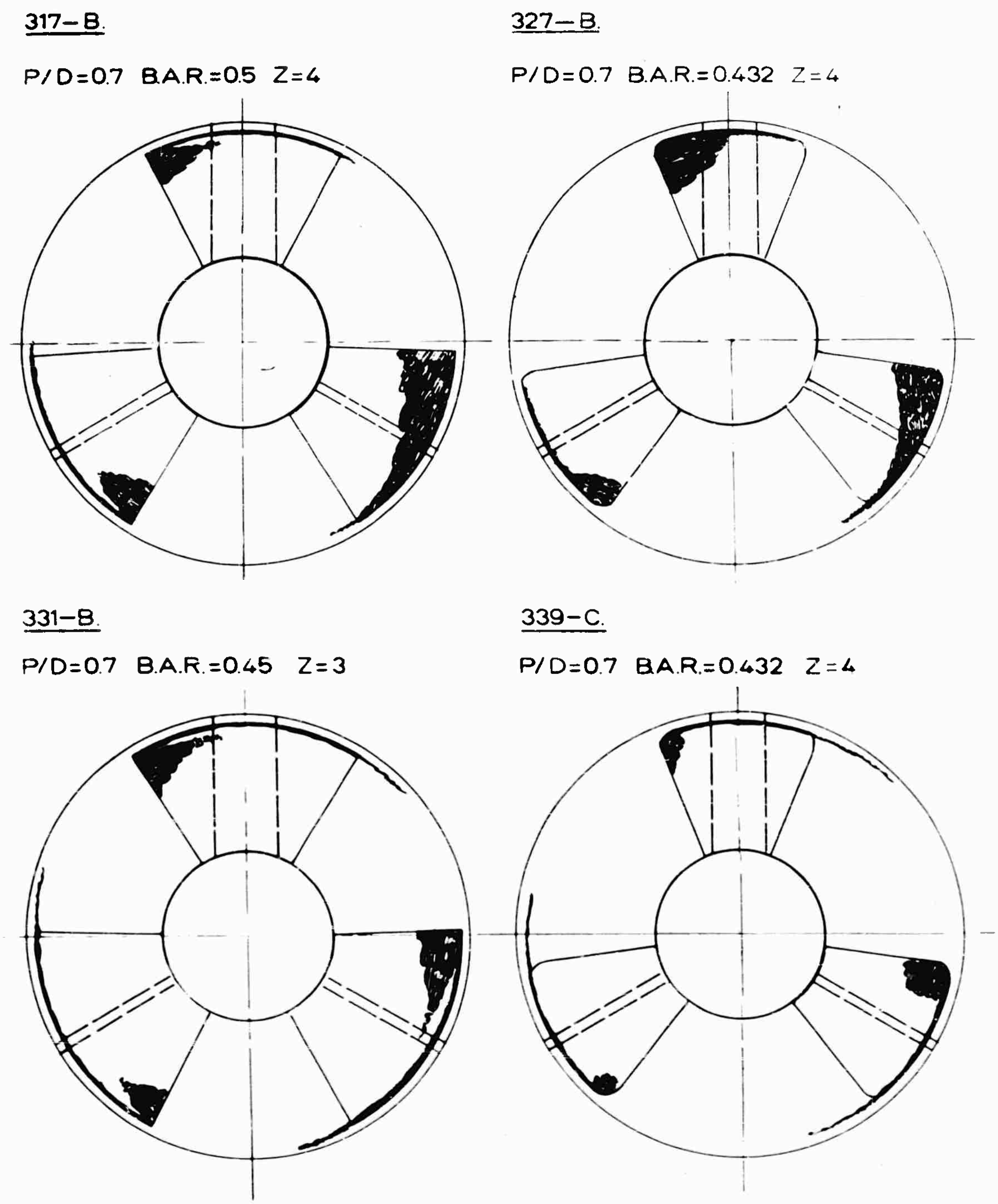


Figure 13 – Cavitation Observation at $\sigma^1 = 2.78$

On each propeller one blade is shown in three various positions.

FULL-SCALE TRIALS

Full-scale test with a how-steering propeller is generally difficult to make quite satisfactorily. The comparatively small thrust in relation to the size of the ship can make influence of wind and current disturbing.

The Danish railway ferry PRINSESSE BENEDIKTE, LOA 111 m, draft 4.50 m, has a 300-hp KaMeWa bow-steering propeller with diameter 1.30 m. The blades are of the 317-B type.

Trial results: 303 DHP $_{\text{metric}}$, 394 rpm and pull 3460 kp which was obtained at a pitch setting $P/D = 0.795$. This gives $K_Q = 0.0330$, $K_T = 0.270$, $\sigma^1 = 3.0$.

By using the above K_Q and K_T, a pitch ratio of $P/D = 0.730$ is ohtained through interpolation in Diagrams 7 and 8. Thus both K_Q and K_T give the same pitch, which indicates that the prediction of power and thrust from the model test was nearly correct, whereas the prediction of pitch was 9 percent low. During the trial the stern of the BENEDIKTE was moored to the quay and the ship extended in about 15 deg out from the quay. It was observed that a circulation was set up around the forebody by the how propeller. Possibly this flow contributed to the higher full-scale pitch.

The propeller was running very vibration-free.

It could have been expected that the thrust and torque should have been influenced by scale effect. The scale effect of friction was calculated to reduce torque by 2–4 percent; the effect of the tip clearance was calculated to increase the thrust by 3 percent,* and, further, the presence of the grid should, according to the model tests, increase the thrust by 1.5–2.0 percent. This would give a total of about 7.5 percent thrust increase as compared with model tests.

During trials of PRINSESSAN CHRISTINA with the same type and size of bow propeller, a thrust of well over 4000 kp was indicated, hut unfortunately the testing equipment was damaged before the trials could be finished.

DIMENSIONING OF PROPELLERS

Diagrams 7 and 8 can be used for determining main data of a bow propeller. A usual prohlem is to find horsepower and shaft speed when a certain thrust and maximum diameter is given. This may be solved by finding a relation hetween K_T and σ^1.

$$\left.\begin{aligned} K_T &= \frac{T}{\rho D^4 n^2} \\ \sigma^1 &= \frac{p_0 - e}{\frac{\rho}{2} D^2 n^2} \end{aligned}\right] \text{ or } K_T = \frac{T}{2\, D^2 (p_0 - e)} \cdot \sigma^1$$

*Schatte, E., "Eine Methode zur Berechnung von Düsen-propellern," Schiff und Hafen, No. 11 (1956).

This straight line can be drawn in Diagram 7. At the intersections with the P/D-curves, σ^1 is read off and the corresponding K_Q is taken from Diagram 8. Thereafter power and shaft speed can easily be calculated.

In the more general case, when only the thrust is given, it can be shown that

$$DHP = \text{const}\ \frac{K_Q}{K_T} \cdot \frac{1}{\sqrt{\sigma^1}}$$

Thus a small K_Q/K_T and a large σ^1 are desired.

A large σ^1 means large diameter and small shaft speed, and it is therefore found practicable to go to about $\sigma^1 = 3.0$. For $\sigma^1 = 3$ it is found:

P/D	0.4	0.5	0.7	0.9
K_Q/K_T	0.106	0.101	0.120	0.143

According to the table, it is therefore not advisable to use larger P/D than around 0.7.

Suitable data would, for example, be:

$$P/D = 0.7,\ K_T = 0.258,\ K_Q = 0.0306,\ \sigma^1 = 2.8$$

With these characteristics it is found that:

$$D = \frac{\sqrt{T}}{46} \quad \text{m}$$

$$N = \frac{24.500}{\sqrt{T}} \quad \text{rpm}$$

$$DHP = 0.885\,T, \text{metric}$$

or

T kp	5000	10.000	20.000
D m	1.535	2.170	3.080
N rpm	347	245	174
DHP	442	885	1770

This paper has dealt with the effect of bow-steering screw propellers at zero ship speed only. However, in many cases steering effect from the bow propeller is wanted also when the ship is proceeding ahead or astern. To investigate this condition, a test series in a model basin is planned.

ACKNOWLEDGMENT

The author wants to express his thanks to Mr. Torsten Stephanson, under whose supervision the experimental work was carried out in the Kristinehamn Cavitation Tunnel.

DISCUSSIONS

A. Silverleaf:

I apologize; I speak without having seen the paper itself and I also speak without the data which I am going to quote.

As the author mentioned, there is an increasing interest in these devices, and recently there have been cases where they have been fitted or at least have been proposed to be fitted both at bow and stern. A little more than a year ago at NPL, we were asked to investigate a proposal which it was intended to fit to a large passenger liner. Our prime purpose was to compare the relative efficiencies (I shall return to this point) of counterrotating and single propellers for the purpose of producing a lateral thrust. Calculations suggested that the single prepeller was bound to be more effective than the counterrotating propeller. Experiments, not surprisingly, confirm this. Our investigations were not made in the tunnel, at least not initially; we were fortunate enough to be able to make them on a fairly large scale. We used propeller models 12 inches in diameter and made a replica of the forebody of the vessel on a correspondingly large scale, which we tested first in a large static water basin and then carried out some further experiments with the propeller in a tube (after all, that is all this is, propellers operating in a tube) in the towing tank.

Now one is interested, basically, in the efficiency of such a device, and this device is essentially that of a propeller operating at zero speed of advance. So some criterion other than normal efficiency must be employed. What one is really interested in, as I see it, is the maximum lateral thrust or lateral force for a given power input. And this quite simply is a ratio of K_T, the thrust coefficient, to the torque coefficient K_Q, raised to the two-thirds power. It is not K_T over K_Q; it is K_T over K_Q to the two-thirds. The surprising thing is that if you examine published propeller systematic data, either for orthodox propellers or alternately for Kaplan type propellers, of which there are now some, you come across the fact that there is an optimum pitch ratio. Quoting from memory, I think that it is around 0.8, at which this criterion of effectiveness is a maximum. And also it is quite easy to derive a relationship between this criterion of effectiveness and a nondimensional ratio in which the power and the diameter are involved. From this, we found it very simple (and I should mention that our measurements broadly confirm this simple analytical approach) to advise on the optimum diameter and running conditions.

Incidentally, this very seriously affects any glib quotation of horsepower per ton thrust, per, say, 100 horsepower; a common figure that is bandied about is 1 ton per 100 horsepower. This can be up or down by a factor of three, quite easily depending on what you can do in the way of diameter relationships for such lateral thrust devices.

I should like to ask a question which may have been answered in the paper; whether, in fact, the author has himself found any such optimum pitch-diameter ratios to exist, and whether he has found any such criterion of effectiveness which varies with the permissible operating conditions.

I was interested in his final comment about the drag of the opening. This is a point that is always raised by a shipowner who comes to us for advice on this topic. Of course I would very strongly agree with Mr. Pehrsson's point that properly designed openings (and I think the shape of the opening is a very important factor here) need not add anything whatever significantly to the drag of the vessel, when proceeding at some speed ahead, without the need for elaborate closing devices which are sometimes suggested and fitted.

The effect of speed ahead is, I think, one that needs to be investigated. Clearly, the flow conditions into the propeller will be affected by this even if the speed ahead is only 1 or 2 knots, which is the condition you are interested in for maneuvering, and in fact, the propeller characteristics may then be considerably altered. Finally, I should like to ask Mr. Pehrsson whether it is to cope with such fluctuating conditions that he advocated the use of controllable pitch propellers, because at the moment I think that they are a slightly unnecessary luxury.

G. R. Stuntz, Jr.:

I'm a little bit in the same position as Mr. Silverleaf in that I haven't seen this paper before. As some of you may know, the recently completed Army dredge MARKHAM, built down at Avondale, is equipped with a device very much like this. We were in a position of doing some model testing for the Army on this boat, and also we did have just a little bit to do with the design of the rig. Like Mr. Silverleaf, we found that a single propeller was more effective than counterrotating propellers. This is possibly due in some degree to the spacing between the propellers and to the fact that they are within a tunnel. Secondly, I am a little surprised at the shape of the tunnel that Mr. Pehrsson described. We recommended to the Army that they not use a sharp-edge junction between the tunnel and the hull surface but rather a very generous radius, so that the inflow of this tunnel had a much easier transition from the external flow to the inside of the tunnel. I believe this might contribute to more stable flow in the way of the propellers. We did, however, use the discontinuity or lip, or step, described in the Swedish work, so that the jet at the outflow end would separate and not follow this rather generous radius. Model tests were done with a large scale model of the bow of the ship. The diameter of the propellers was about 9 inches on the model.

Unfortunately, we didn't get into any work with guide vanes; in fact, in the model work we had no grid whatever over the entrance and exit, and I have been interested to note the ship was built with such a grid. We don't have any full-scale trial data, so I don't know how that turned out. One other point in which I support Mr. Silverleaf is that the controllable pitch propeller does seem to me to be somewhat of a luxury. It requires a very much larger hub, I suspect, than is required merely to house a right-angle drive. And certainly this large hub inside a tunnel would seem to be a disadvantage.

R. N. NEWTON:

I, too, haven't seen the paper until today and therefore base my comments on the other discussers. I would like to see the title of the paper put another way, such as, "Are the days of tugs numbered?" Quite frankly, I think they are and should be. To anybody who has been in the Navy actively, as I have, nothing is more annoying as they go up the Clyde in a battleship with four wretched tugs fussing around, pushing you this way and that, with strong language from the bridge. Then the QUEEN MARY comes up to her docks and pokes her nose in and everything is all set. But there is a need here for tug propulsion units. There is no doubt about it; if one can do away with tugs it facilitates the operation an awful lot.

One question I would like to ask, that can be answered with a bit of discussion, has to do with the clearance between the faired tube and the blade tips, and did it have an effect on tip vortex cavitation?

I took note of the discussion about the relative merits of the counterrotating propeller in a tube, and here I would agree with Mr. Silverleaf. I think that one great use of the counterrotating propeller is when it is placed behind the ship, because you can take more advantage of the wake of the ship, which is one reason why the counterrotating propeller behind the ship is more efficient than the single propeller.

But in looking for another point, about the ehp of the opening. If you can make the propeller smaller by a counterrotating propeller you make the opening smaller and you undoubtedly effect more economy, from the point of view that you do not use so much ehp. The ehp is not negligible. We at Haslar were investigating the so-called ducted propellers and we do take into account the loss of ehp as it left the pipe. In a destroyer at 30 knots it is of the order of 300 hp. These are not small openings, of course, and I think the counterrotating propeller would reduce that quite a bit.

H. E. Saunders:

I have read the paper once, and I am confused as to how to make the distinction between propeller thrust and thrust on the ship: the sort of thrust that you are endeavoring to produce to turn the ship, or to move it sideways? You have only one symbol T, and I am uncertain as to whether that applies to the thrust that is being developed by the propeller only or the thrust on the entire assembly as far as the ship is concerned.

S. Bindel:

Regarding the drag of the hole, we carried out some tests on cargo ships and ferries and we found that the increase of ship resistance due to bow tunnel was not more than 2 percent; our results confirm, for relatively slow ships, those given by Mr. Pehrsson.

Author's Closure:

I was very interested to hear Mr. Silverleaf's remarks regarding two counterrotating propellers. One thing which my company has found, is that both propellers of a counterrotating pair should not have the same diameter. This was verified by experiments on torpedo propellers made in our cavitation tunnels about 15 years ago. It was demonstrated that, if the diameters are the same, the flow into the downstream propeller is disturbed by the tip vortex of the forward propeller.

I both disagree and agree with Mr. Silverleaf on the point of comparing the efficiency of the propeller by K_T/K_Q or $K_T/K_Q^{2/3}$. It is quite correct when you base it on a given horsepower to compare on $K_T/K_Q^{2/3}$, but if you compare for a given thrust then you should use K_Q/K_T.

There was some discussion on the shape of the openings; we have found that the most effective is a sharp opening at the forward half of the diameter and a rounded opening at the following part of the tunnel.

Then there is the question–why a controllable pitch propeller? I don't think it is a luxury here but the reverse. It is so much cheaper with an electrical system to use a non-reversible constant speed motor. Therefore, the total cost of the installation is much less using a controllable pitch propeller.

I would agree with Mr. Stuntz that counterrotating propellers can be advantageous in some cases, and especially where a small draft forbids the use of a large tunnel opening.

In reply to Captain Saunders, K_T means the total thrust delivered by the propeller and the tunnel, as indicated in the upper part of Figures 7, 9, and 11. Thus this coefficient indicates how much thrust is delivered to change the course of the ship.

MODEL EXPERIMENTS ABOUT THE MANEUVERABILITY AND TURNING OF SHIPS

by

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ABSTRACT

This paper is composed of three parts as follows:

Part 1: Self-propelled model experiments about the maneuverability of ships.

Part 2: Effects of screw propeller on the character of the rudder.

Part 3: Effects of the depth of submergence of the rudder on its performance.

In Part 1, the author investigated generally about the optimum rudder area, the maximum rudder angle, and the effect of the C_b of the ship's hull on the maneuverability of ships.

In Part 2, he showed that the optimum rudder area is decided mostly from the ratio of C_N/β, the ratio of the normal pressure coefficient to the aspect ratio, independently from the ship's hull when the depth of the rudder has to be kept constant as is usual. He also presented the design data for the rudders working behind propellers.

In Part 3, he made clear that by decreasing the depth of the submergence of the rudder, the normal pressure decreases, nevertheless the rudder torque stays at the same level, at large angle of deviation, because of the backward shift of the position of the center of pressure, which cancels out the decrease of the normal pressure. He showed that this is the reason why the design data aquired from the deeply immersed rudder can practically be applied for the rudder of actual ships.

PART 1

SELF-PROPELLED MODEL EXPERIMENTS ABOUT THE MANEUVERABILITY OF SHIPS

1.1 INTRODUCTION

There is very little scientific information and technical data available about the maneuverability and turning of ships, and we still have many unsolved problems about these performances. Design data for the rudder is insufficient; for example, we have difficulty finding reliable data to decide the rudder area that will give the best maneuverability and turning.

Here the author performed a series of self-propelled model experiments for the purpose of getting practical design data to decide the best rudder area. This is one phase of a rather comprehensive series of model turning experiments, and the general results of this study are reported here in Part 1.

1.2 EXPERIMENTAL SETUP

1.2.1 MODEL HULL, RUDDER, AND SCREW PROPELLER

We used three model hulls which have the same length/width ratio (L/B) and width/draft ratio (B/d) of conventional ships, and which have three different block coefficients, 0.6, 0.7, and 0.8. We chose these coefficients because they represent, respectively, a high-speed merchant ship, a general cargo ship, and a super-large-sized oil tanker. The principal dimensions of the model hull are all 2.50 m in length, 0.3425 m in width, and 0.137 m in draft, and the ratios are $L/B = 7.30$ and $B/d = 2.50$. Bow and buttock lines and the forms of stem and stern of these model hulls are shown in Figure 1, as listed in Table 1. For these tests, we did not put the bilge keels on the models. The effects of bilge keels should be studied separately.

TABLE 1

Kind of Model Hulls Used

Model	Kind	Block Coefficient	Body Plans
A	High-speed merchant ship	0.60	Figure 1
B	General cargo ship	0.70	
C	Super-large-sized tanker	0.80	

The rudder area ratios $A/L \times d$ of five rudder models are respectively 1/80, 1/60, 1/50, 1/40, and 1/35. The five rudders are all rectangles which have the same length (span), 9.8 cm, but the length of the chord differs for each rudder. As a result, the aspect ratios differ; namely, 2.24, 1.68, 1.40, 1.12, and 0.98.

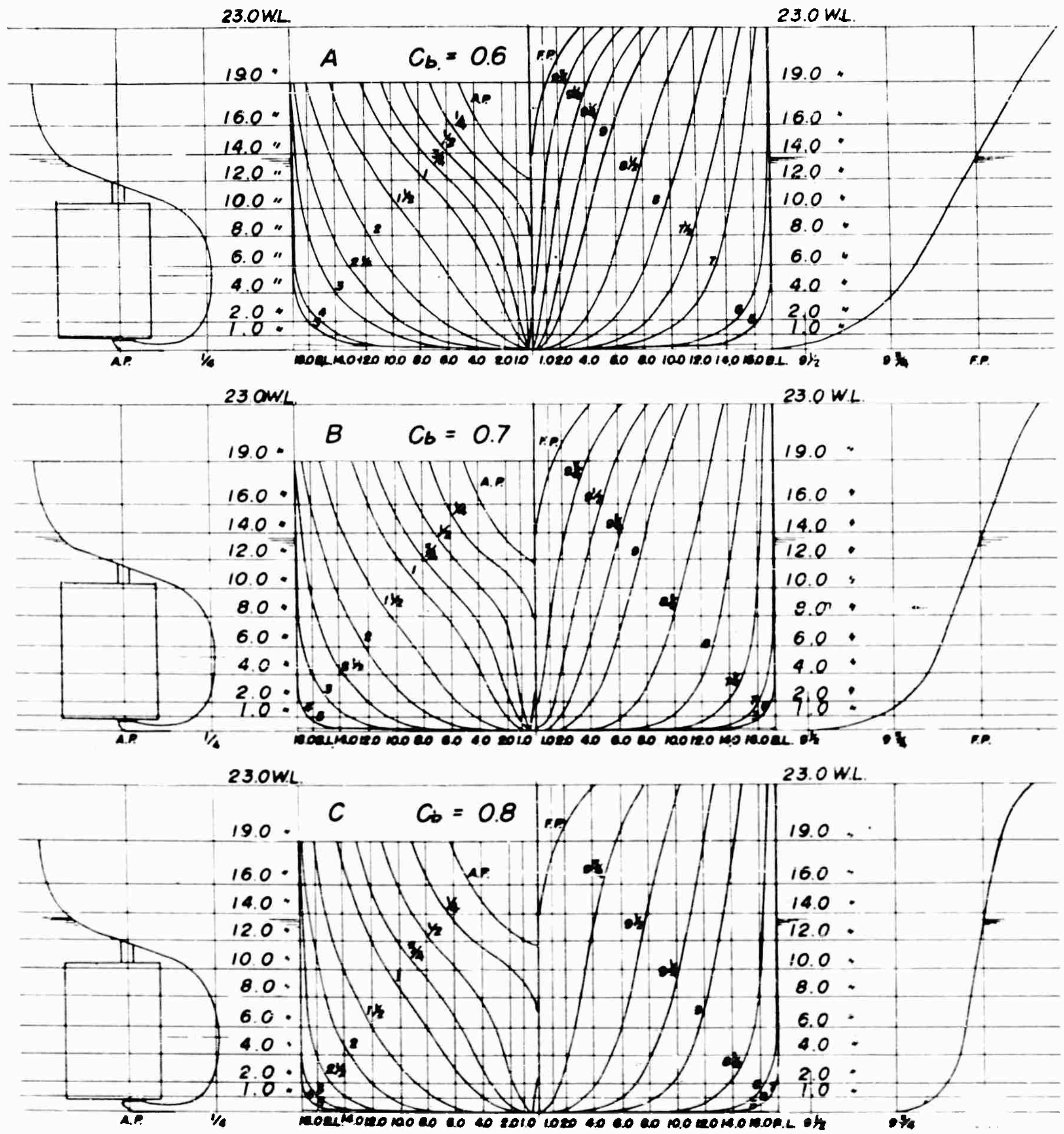


Figure 1 – Body Plans

We adopted NACA 0018 airfoil for the section form, putting its shaft position at 33 percent from the leading edge. The screw propellers are the SENPAKU SHIKENJYO No. A4-40 type, which are 10 cm in diameter, and have three different pitch ratios, 0.7, 0.8, and 0.9. All propellers are right-handed.

TABLE 2

Principal Dimensions of Propeller

Diameter	100 mm
Boss ratio	0.250
Pitch ratio (constant)	0.7, 0.8, 0.9
Expanded area ratio	0.40
Maximum blade width ratio	0.242
Blade thickness ratio	0.045
Rake	10 deg - 18 ft
Number of blades	4
Turning direction	Right-handed

1.2.2 PROPULSION MOTOR

A 1/12-hp, 24-v d-c shunt motor was used as the driving motor for the screw propeller. 30-v, 6-amp-hr storage batteries were used to supply the electric sources to this motor and other apparatus in the model.

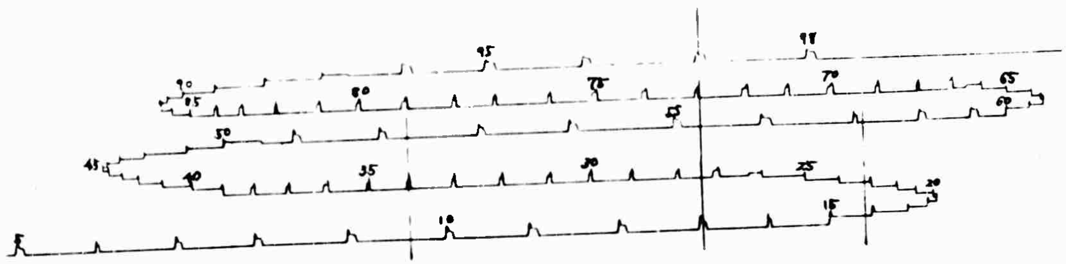
1.2.3 STEERING MECHANISM

The steering mechanism is such that a spring supplies the power, and by adjusting its windings we can steer the rudder to the pre-set angle in a given time.

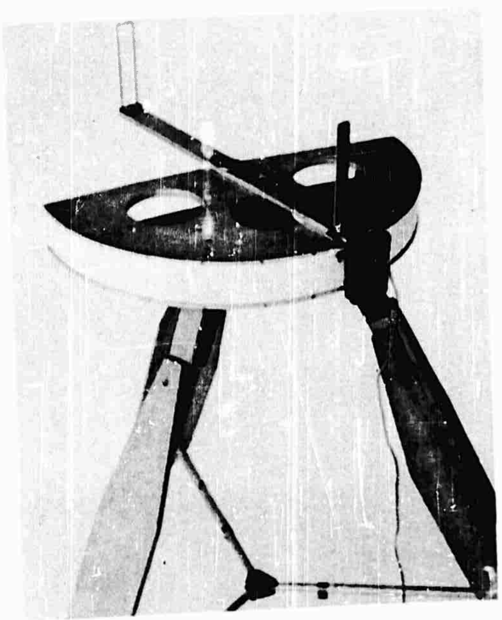
1.2.4 TRANSIT FOR PURSUING THE PATH

A transit, which was designed by the Ship Performance Division, Transportation Technical Research Institute (TTRI), was used to track the model. Picture 1 shows its appearance, and Picture 2 shows an example of the measured record.

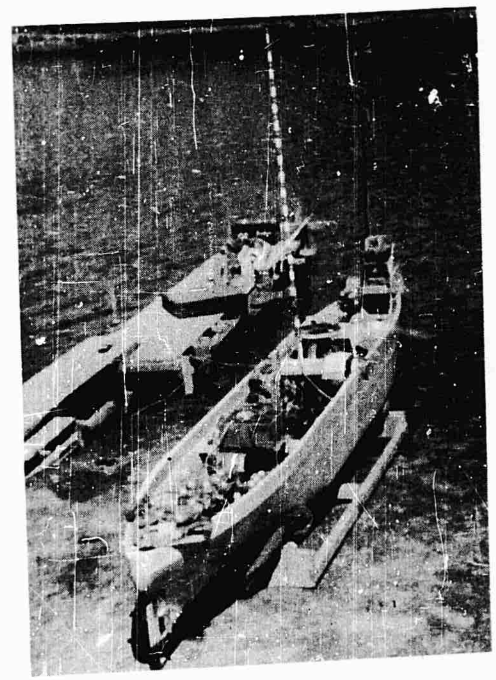
While the observer pursues the position of the model continuously on the transit, the direction is recorded automatically at every second or at every other second on the drum. We used two transits in this series of tests. The model of block coefficient 0.70 is shown in Picture 3, loaded with all of the above-mentioned experimental devices and apparatus.



Picture 2 – An Example of the Measured Record



Picture 1 – Transit for Pursuing the Path



Picture 3 – One of the Model Ships Used

1.2.5 MISCELLANEOUS APPARATUS

Also used in this experiment were:

a) Gyroscope for course angle meter,

b) Wireless controlling device.

1.3 METHOD OF EXPERIMENT

When the model reaches the steady-state condition, the pre-set programmer starts and the number of revolutions of the main shaft is measured first; then the rudder is steered and the model starts to turn.

Besides, starting from the time the model makes a straight advance, the course angle is continuously recorded by gyro indicator and cine camera and the trace of the model is followed by transits on the shore. Finishing the required measurements, the model is stopped by the signal sent by the wireless controlling device.

By this method the path of the model is found at every second (or at every other second). This gives us the advance, transfer, turning circle, turning diameter, and speed of the ship, together with the number of revolutions of the main shaft, and the course angle at every second. From this, we can find, further, the speed of the change of the course angle and the drift angle.

These experiments were held last March and May on a reservoir in the northern suburbs of Numazu, Shizuoka-Prefecture and at the experimental basin in Meguro-Ku, Tokyo.

1.4 KINDS OF EXPERIMENTS

We used three model hulls, five rudders, and three screws in the experiment; rudder angle, speed, trim, displacement, and turning direction of the models were changed, as shown in Table 3. We examined the effect of these variations upon the maneuverability of the ship; for example, on the turning diameter, advance, transfer of turning, reduction of speed, and so on. The number of runs in this series was more than 200.

TABLE 3

Kinds of Experiments

Helm angle	10 deg, 20 deg, 30 deg, 35 deg, 40 deg, 45 deg
Speed (Froude No.)	0.09 – 0.33
Trim	Aft trim, even keel, fore trim
Displacement	Light- and full-loaded condition
Turning direction	To port and to starboard

1.5 RESULTS OF EXPERIMENTS

The model drifts more or less in a certain direction because of the wind force while it is running. This means that the model is much more affected by wind of the same speed than the actual ship is, because the advance speed of the model is equal to $1/\sqrt{\lambda}$ (λ is scale ratio) times that of the actual ship, by Froude's Law.

To correct or eliminate such a wind effect, measurements were made until the model completed the steady turning circle twice.

The direction and the distance of drift during one turning circle are determined from the trace of the turning circle. So the effect of the wind per unit time can be found if we assume that the direction and the distance of drift remain constant for a short period of time. By this method we can modify the trace of the model, eliminating the wind effect. This simple method gives almost satisfactory results, as shown in the example in Figure 2, where a comparatively large drift was eliminated by this method.

From more than 200 turning traces acquired by the experiments, several representative examples about the model of $C_b = 0.7$ are shown in Figure 3, corrected for wind effects.

The figure numbers and corresponding experimental data are shown in Table 4.

TABLE 4

No. of Figure	Rudder Area Ratio	Block Coefficient	Propeller Pitch Ratio	Trim	Displacement	Direction of Turning
Figure 5.1	1/80	0.70	0.80	Even keel	Full load	Port side
Figure 5.2	1/60	↓	↓	↓	↓	↓
Figure 5.3	1/50					
Figure 5.4	1/40					
Figure 5.5	1/35					

1.6 CORRELATION BETWEEN MANEUVERABILITY OF MODEL AND ACTUAL SHIP

To apply the experimental results obtained with the self-propelled model to the full-sized ship, we must examine the correlation between the model and the ship in maneuverability. We note the following facts:

1. In the model experiment, the Reynolds number is smaller than that of the ship; therefore, the characteristics of the rudder of the model differ from that of the ship.

2. It is possible for the screw of the ship to cavitate; however, with the model that very seldom happens.

(Text continued on page 63.)

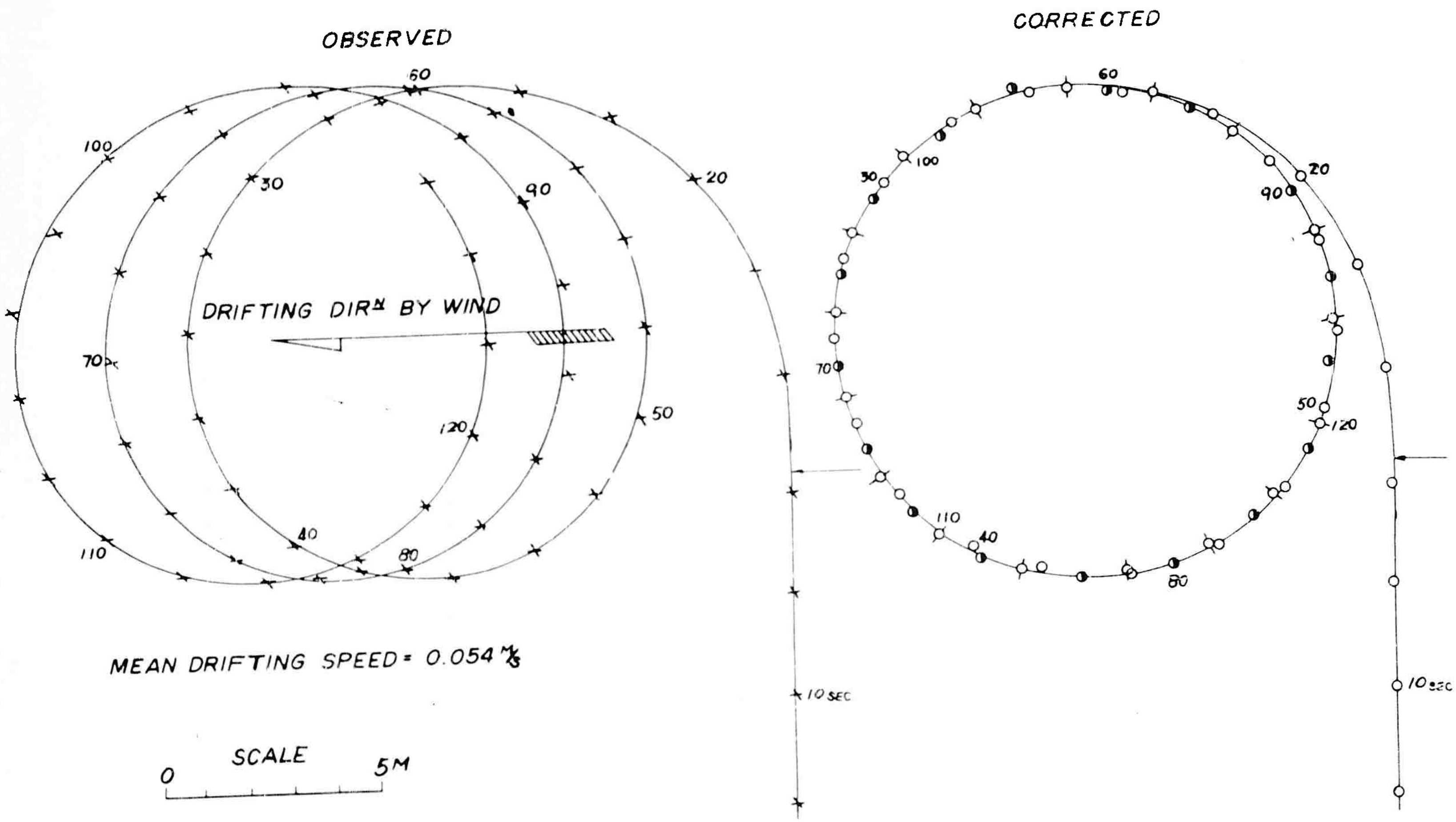


Figure 2 – An Example of Turning Path Observed

Figure 3 – Turning Path After Steering

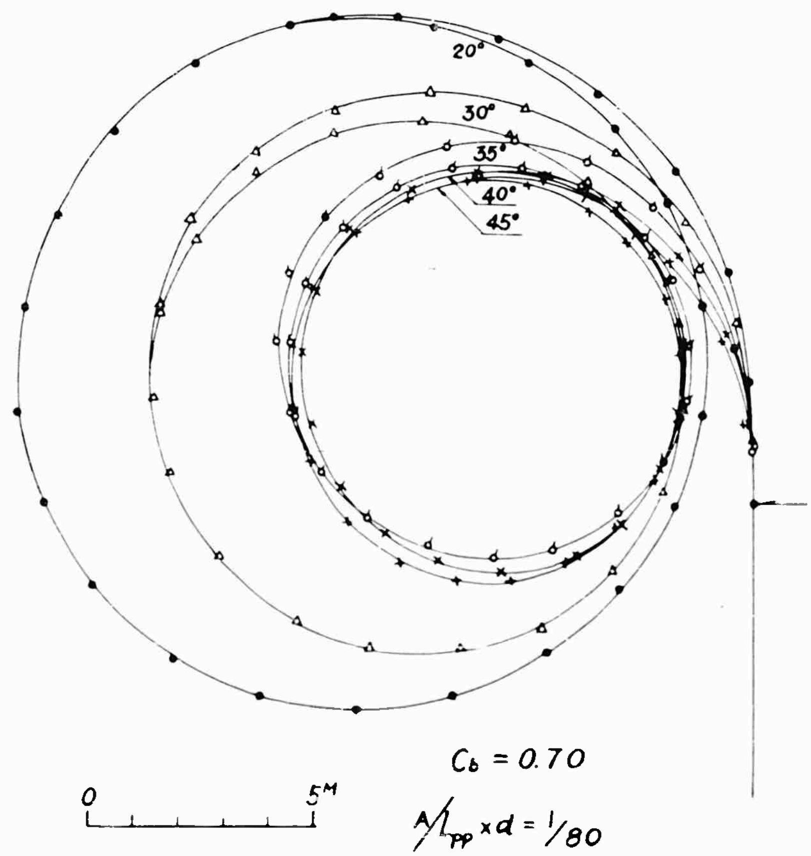


Figure 3-1

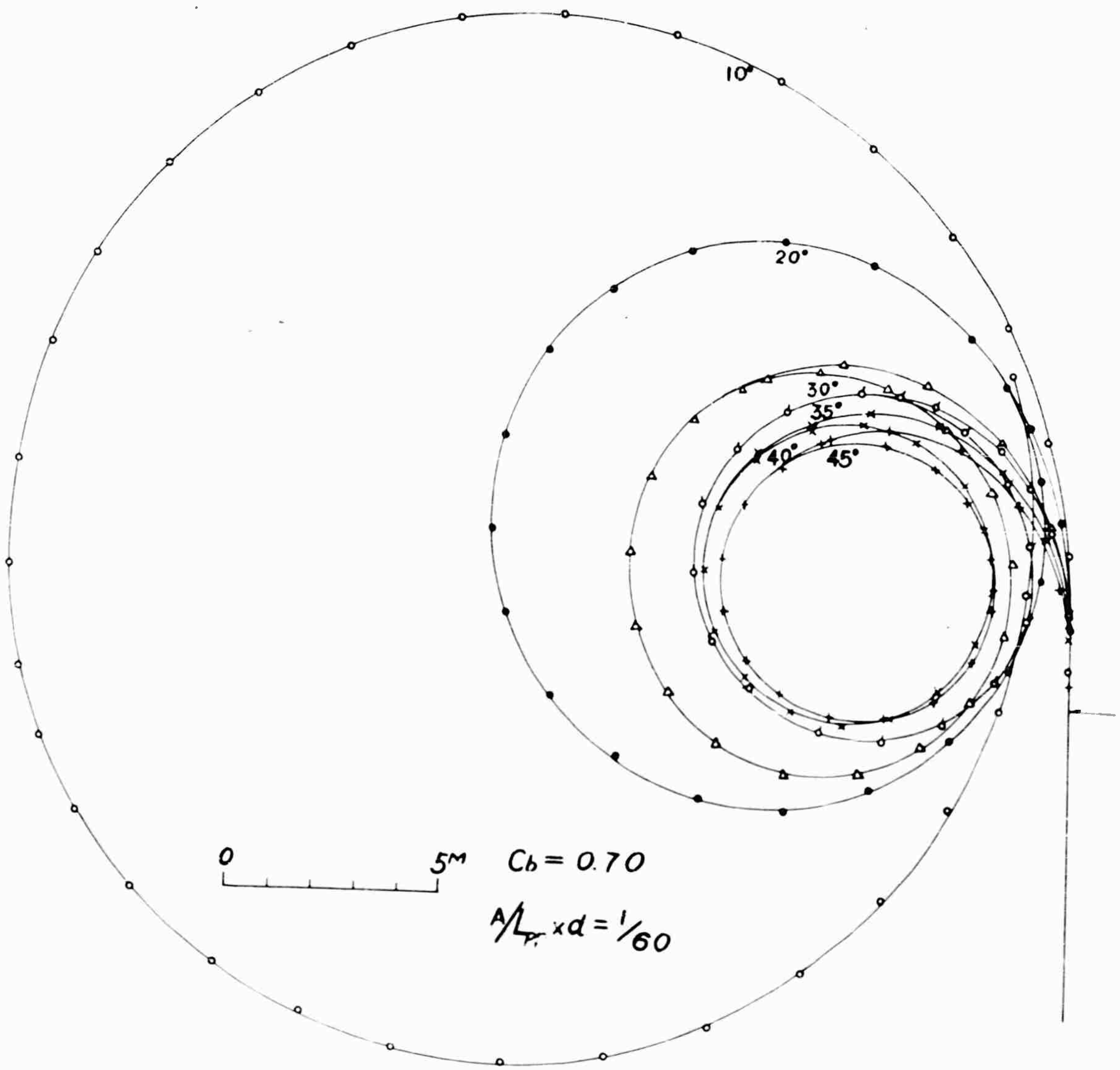


Figure 3-2

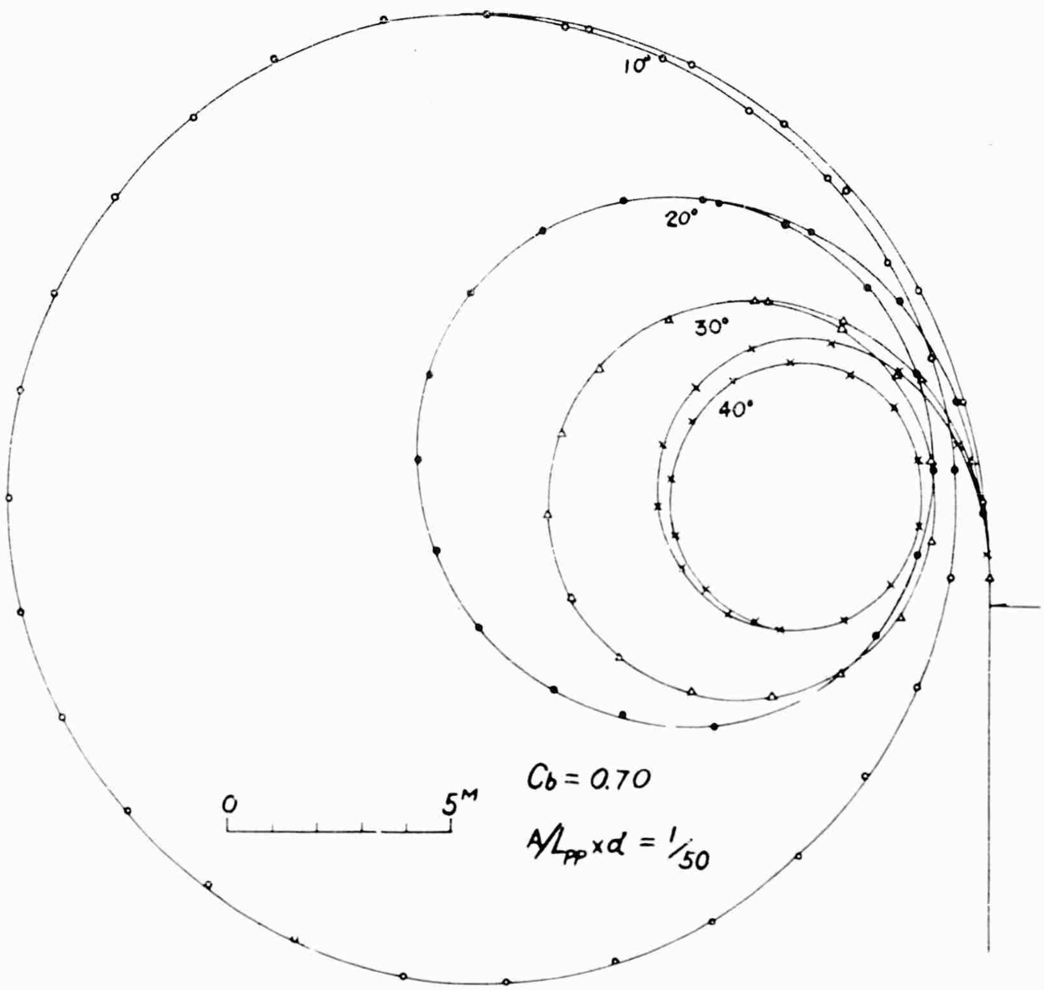


Figure 3-3

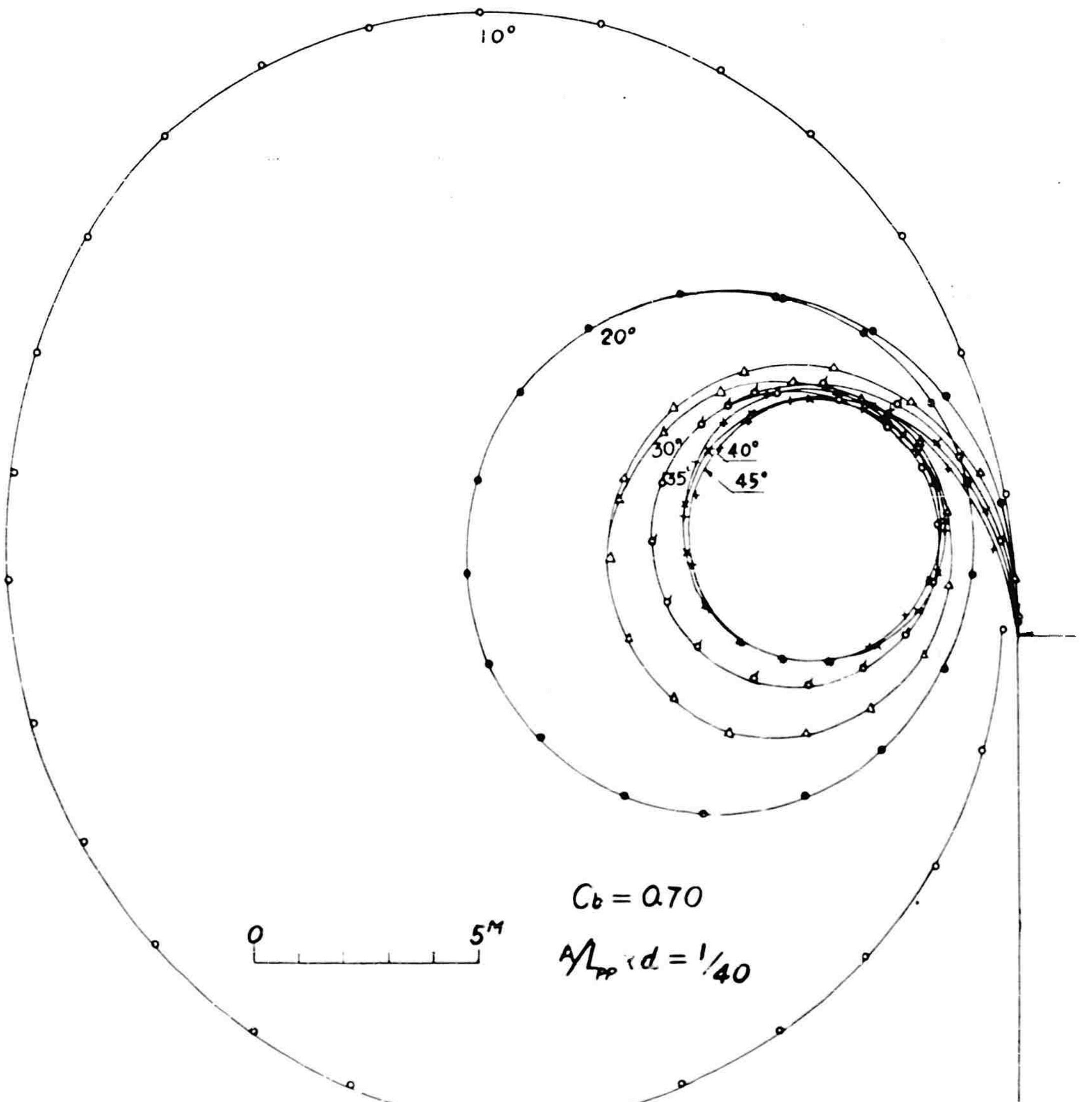


Figure 3-4

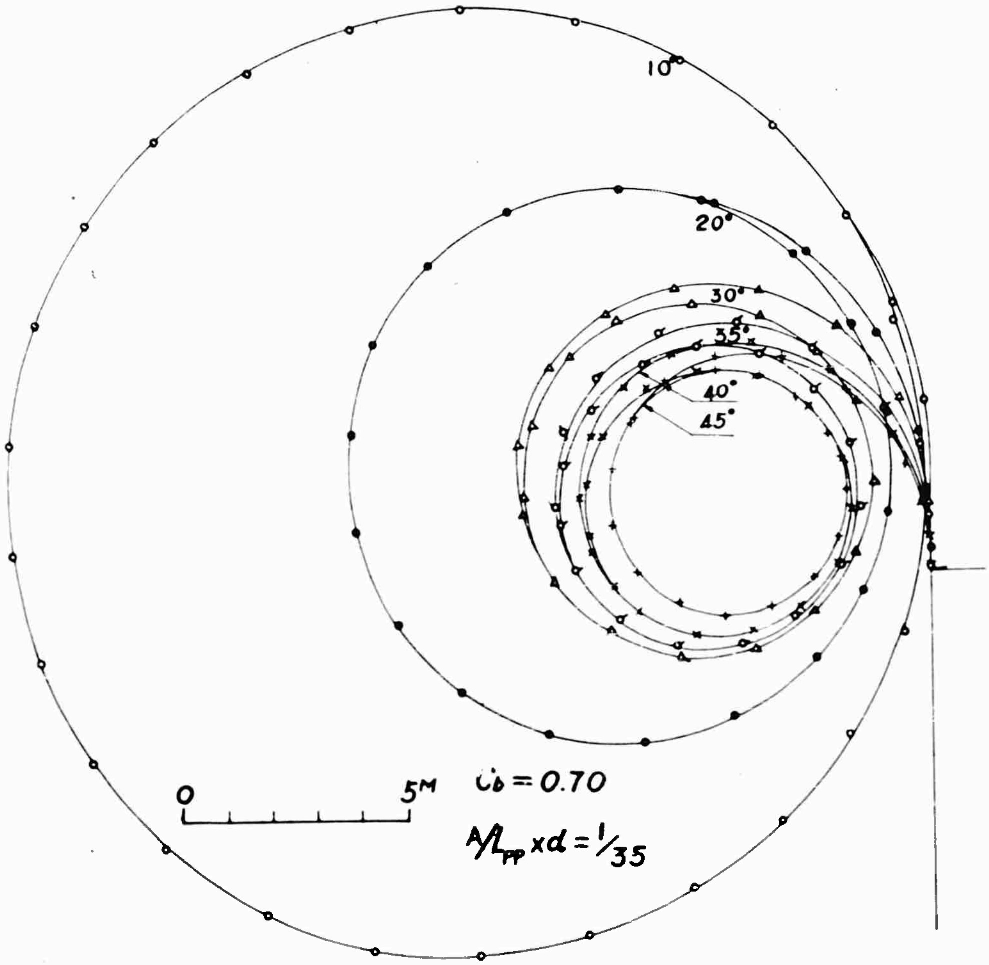


Figure 3-5

3. As the Weber number of the model rudder is considerably different from that of the actual rudder, the phenomenon of air draw in the model is considerably different from that of the actual ship. However, the results of the turning experiment with the model agreed fairly well with those of the ship trial. Brard, Hewins and Roop, and Pitre, have already reported this fact; also, the test results on models and ships carried out by the Ship Performance Division, TTRI, showed good agreement.

Experiments were carried out on models and ships designated X and Y, and comparisons of those models and ships are shown in Figure 4-1. Ship X is a twin-screw twin-rudder ship

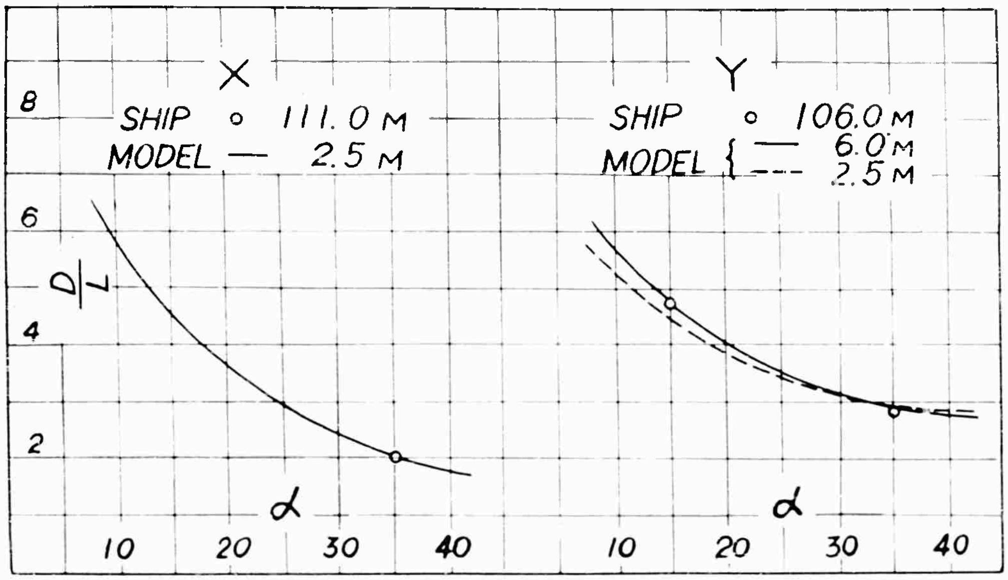


Figure 4-1 – Comparisons of D/L between Ship and Models

111 m long, 17.40 m wide, 6.80 m deep, and 4.78 m in draft at the test condition, and all rudders are just behind the screws. The ship trial was carried out at 14.4-knot speed and 35 deg in the helm angle. The model was 2.5 m long, and the model tests were held at a speed of 1.12 m/s (corresponding to 14.5 knots for the ship) and at the helm angles of 10 deg, 20 deg, 30 deg, and 35 deg. As shown in the figures, the model test results agreed sufficiently well with those of the actual ship. Ship Y is also a twin-screw twin-rudder ship 106.0 m long, 10.5 m wide, and 3.67 m in draft, and the rudders are deviated a little from the center line of the propeller stream. Two models 2.5 m and 6 m long were used for the experiments in this case. The 2.5-m model was used first, but in order to investigate the scale effect and to increase the accuracy of the experiment the 6-m model was used later. Test results with the 6-m model agree well with the results of the sea trial with the actual ship. Furthermore, Figure 4-2 shows a comparison of the experimental results of this test with the results of sea trials of scores of ships recently built in Japan. The results of the sea trial are plotted by

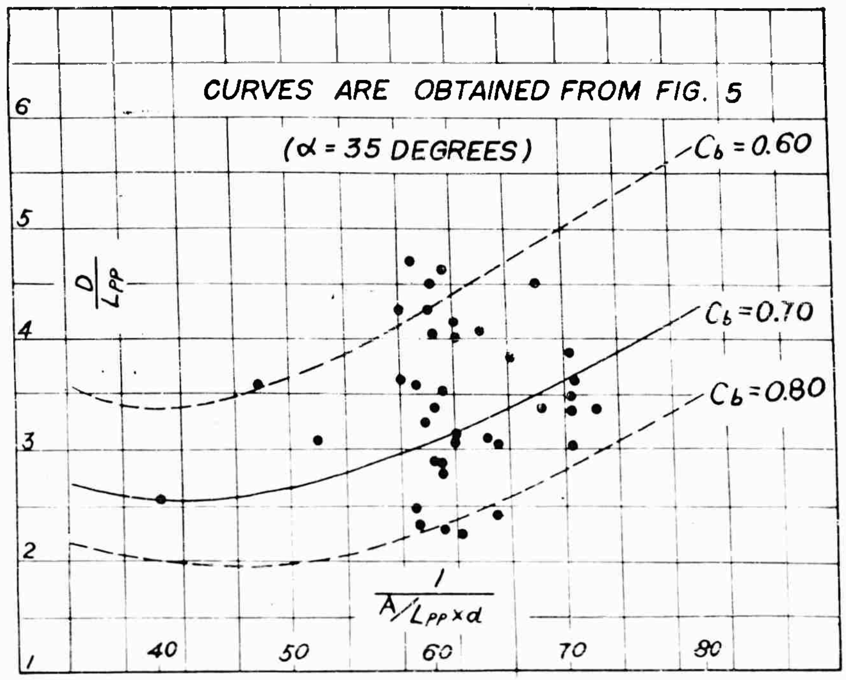


Figure 4-2 – Values of Turning Diameter Obtained from Trials at Sea

circles, and the model test results are shown by the solid and dotted lines. Block coefficients of the ships range from 0.65 to 0.75, but the results were modified to the case of block coefficient 0.70 with even keel condition, in order to make the comparison easier. For this calculation, experimental results obtained with models, described in the following section, were applied. There still remains the influence of the differences in L/B and B/d and the differences in the wind and tide effects, on these results, that make the plots scatter over a wide range. As a whole, however, the fact that the experimental results plot closely to each other, having the lines of the results of model test of $C_b = 0.7$ as their average line, means that the similarity is well kept between models and actual ships. In the model test, there exist the scale effects in its friction; namely, the corresponding frictional resistance and the wake of the model are relatively larger than those of the ship. Accordingly, at the corresponding speed, the screw needs to produce a larger thrust than the corresponding thrust of the ship, if the model is propelled without friction correction. However, as the effect of the difference of wake and the effect of neglecting the friction correction cancel out each other, so the stream to the rudder remains almost similar between model and ship. In short, practically, we can say that the correlation between model and ship shows a good agreement as mentioned above. This is true, however, only in the range of the case where cavitation and air drawing never occur in the screw of the ship.

1.7 DISCUSSIONS OF RESULTS OF EXPERIMENT

1.7.1 RUDDER AREA

Variations of turning diameter with the increase of the rudder area ratio in three models, A, B, and C, are indicated in Figure 5. Turning diameter decreases considerably with the increase of the rudder area ratio, in the range of, say, 1/80 to approximately 1/50. It should be noted, however, that the turning ability is not improved any more when the rudder area ratio is increased beyond 1/50. To obtain the best turning ability, we should have the optimum value of rudder area ratio, say, approximately 1/45. Of course it depends on the ship form, as is shown in Figure 5; it is nearly 1/40 in A, 1/45 in B, and 1/50 in C.

Relations between rudder area ratio and the advance or transfer are shown, respectively, in Figures 6 and 7. As for the advance, there is no best rudder area ratio, as was found for the turning diameter, and it keeps decreasing gradually with the increase of the rudder area. Relation between the transfer and the rudder area is, on the contrary, almost the same as that between the turning diameter and the rudder area. The advance and transfer might be considered as the indices of maneuverability in transient state, while the turning diameter is as that in the steady state. Since the ship movements in transient state are significant, for example, for preventing collisions, we are now investigating this problem separately.

Based on the model experiments with towed models, Gawn in 1943 described the relations between the rudder area ratio and the turning diameter, the advance, and the transfer. The results obtained from our self-propelled model experiments are shown in Figures 5 to 7, which give further information about this problem and are applicable for practical purposes.

1.7.2 RUDDER ANGLE

Turning diameter, advance, and transfer gradually decrease, in general, as the rudder angle increases to a certain angle. This tendency is recognized not only in rudder angle above 35 deg, but also in large rudder angle around 45 deg (Figure 8). Gawn also reported the same result with respect to this point, and here further details are clearly obtained by the self-propelled model experiments.

The effectiveness of this large rudder angle can be explained by the rather small effective angle of incidence to the rudder and also by the fact that the stall very seldom occurs because of the confused aftstream of the propeller. If we neglect the effect of propeller stream, subtracting the drift angle at the rudder from the rudder angle, we can obtain the effective rudder angle. As was found clearly in this experiment, the drift angle at midship reaches to 10 deg to 17 deg when the rudder is steered to 35 deg. Then the drift angle at the rudder can be considered to be more than about 20 deg. Therefore, the effective angle of incidence to the rudder is still less, and at this rather small rudder angle of attack, as the rudder angle increases, the normal pressure still tends to increase. There are the reasons why the steering by such a large helm angle as 45 deg is still effective for improving the

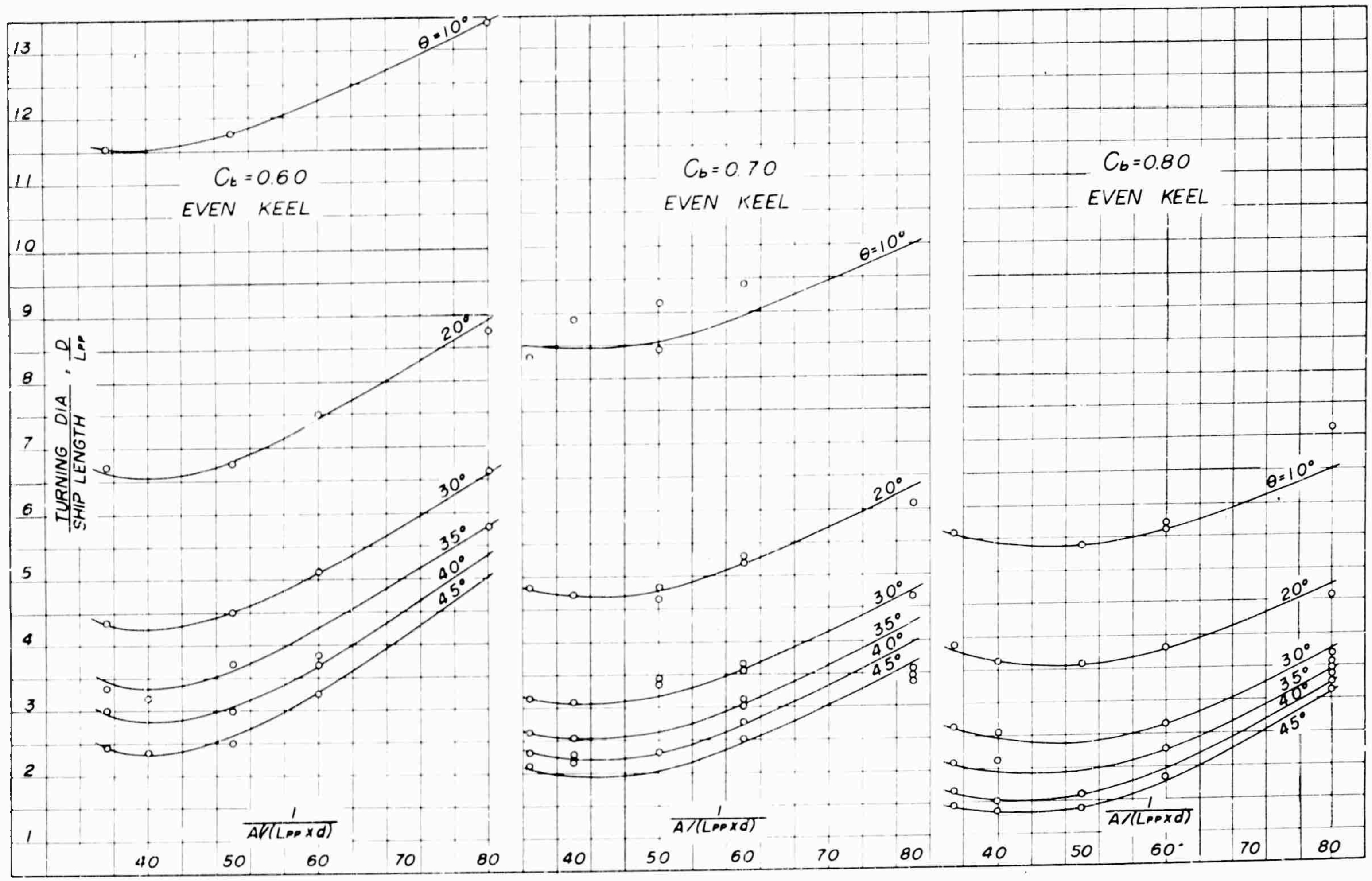


Figure 5 – Effect of Rudder Area upon Turning Diameter at Full Load

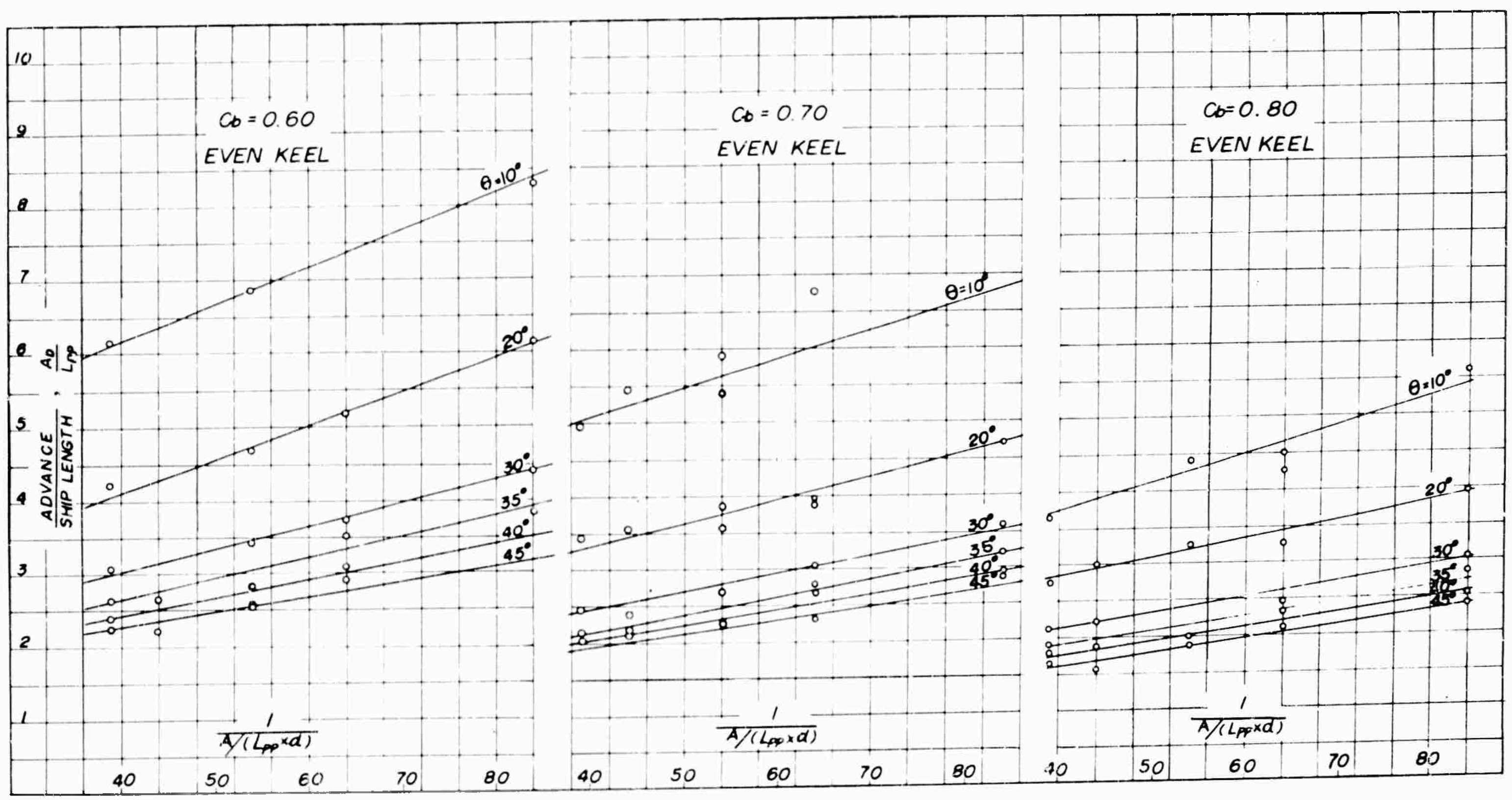


Figure 6 – Effect of Rudder Area upon "Advance" at Full Load

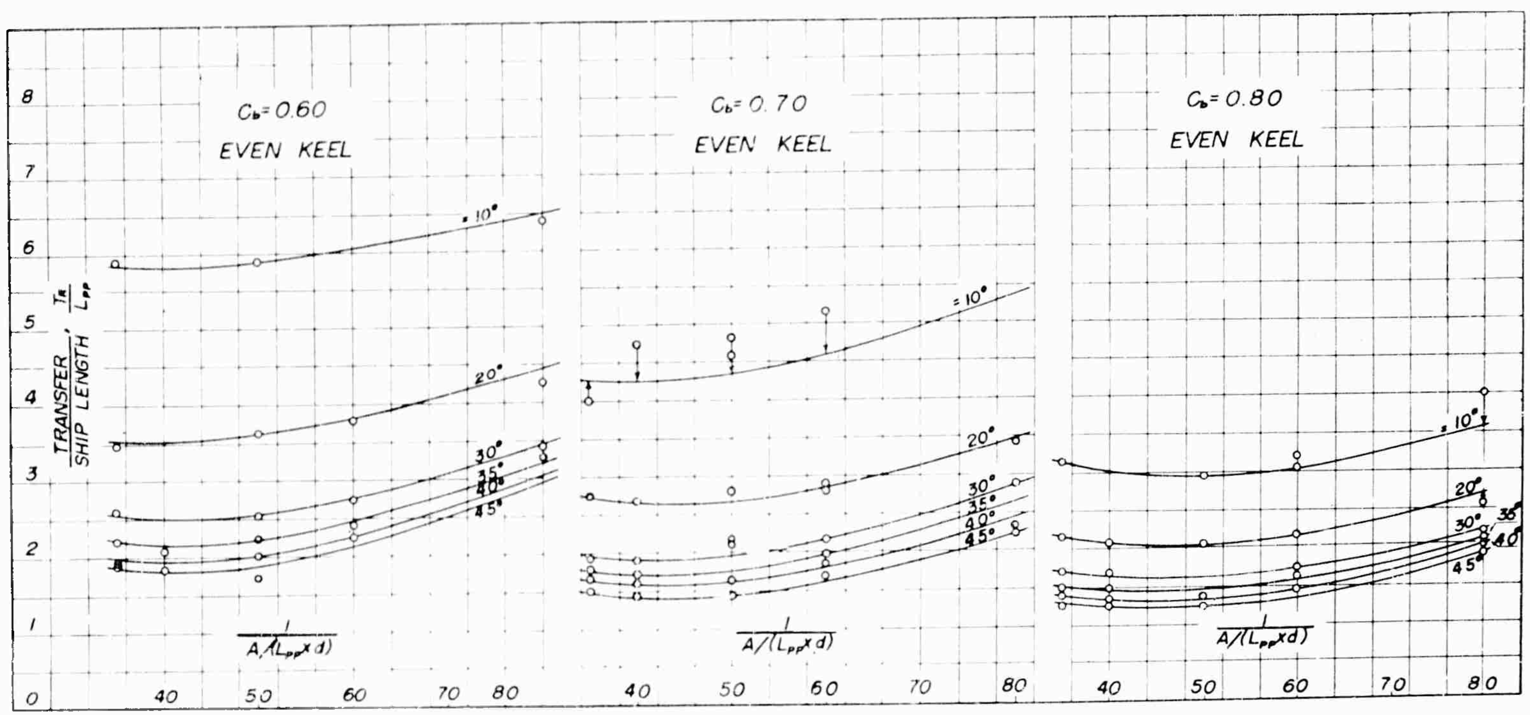


Figure 7 – Effect of Rudder Area upon "Transfer"

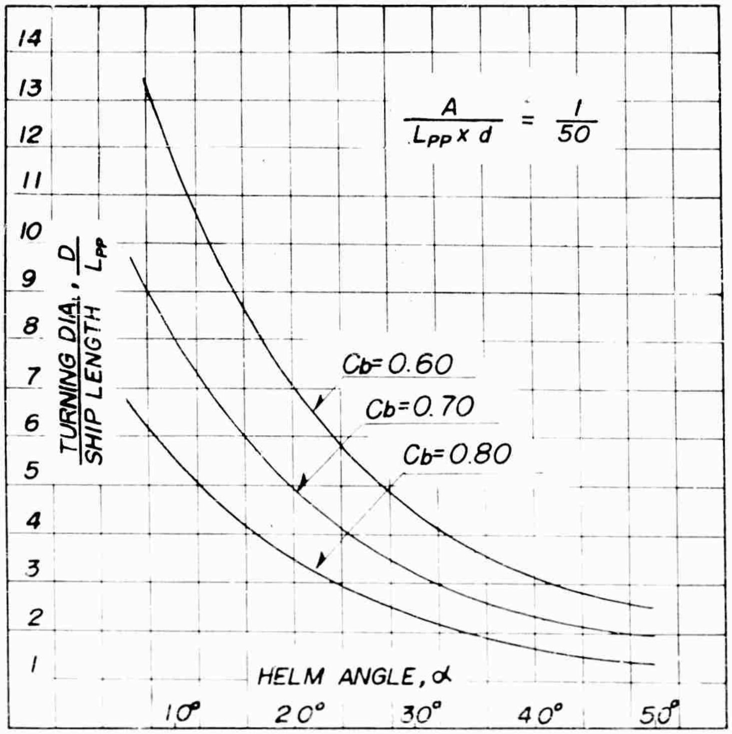


Figure 8 – Effect of Helm Angle

turning ability. Conventionally, the maximum helm angle in the ships is limited to 35 deg. As is shown in Figures 5 and 8, however, in the case of a ship, which especially needs a good turning ability, it is more effective to increase the maximum helm angle than to increase the rudder area only. In this experiment, D/L of the Ship B ($C_b = 0.70$) is found to be 2.70 when the rudder angle is 35 deg and the rudder area ratio is 1/50, the turning diameter is not reduced remarkably by the increase of the rudder area ratio until it reaches 1/35. When the rudder is steered to 45 deg, however, leaving the rudder area ratio 1/50, D/L becomes 2.10. That is, it is improved as much as 22 percent. When the rudder angle becomes more than 50 deg, the turning ability does not improve any more by the increase of the helm angle, as is expected, and is shown in Figure 8.

1.7.3 SHIP FORM AND TURNING

As the elements of the ship form which affect the turning ability, we can count many factors such as the principal dimensions, L/B, B/d, block coefficient, cut-up of the stem and stern, and so on. Here, the relation between the turning diameter and the block coefficient is shown in Figure 9 for the cases when the principal dimensions and rudder are the same, and as are shown in Figure 1, the profiles of the stern are different from each other with the block coefficients. Variation of the turning ability with the change of the ship form is remarkable in this case where the principal dimension and rudder stay the same and differ only in the

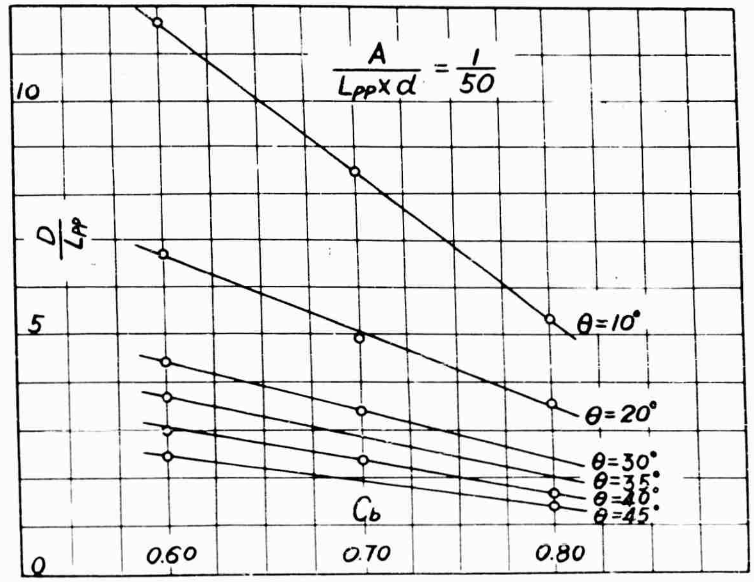


Figure 9 – Effect of Ships' Fullness upon Turning Diameter

block coefficient. Figure 9 shows, for example, that the turning diameter of the ship of block coefficient 0.60 is about twice that of the ship of block coefficient 0.80. Of course, the reason for this big difference is in the difference in the turning moment of the ship caused by the difference of the block coefficient. Besides that, however, the slip ratio will increase in the ship as the block coefficient increases, and this makes the rudder in the propeller stream act more efficiently. In short, when the rudder area is determined to obtain a good maneuverability, the block coefficient should be considered in connection with it.

The lines of Models A, B, and C are chosen to represent the high-speed liner, the general cargo ship, and the super-large-sized tanker, respectively. In these ships, the relations between the turning diameters and the block coefficients are nearly linear. Figure 8 shows the case where the rudder area ratio is 1/50.

1.7.4 EFFECT OF TRIM

The variations of the turning diameter, advance, and transfer, which change with the change of the trim are shown in Figure 10. These are obtained with the Models A, B, and C, with the rudder of the rudder area ratio 1/50, keeping the displacement the same and changing the trim only. Turning ability is very much affected by the change of the trim, as is expected. From Figure 10 we can find, for example, that if we increase the trim by the stern by 1 percent the turning diameter increases by about 10 percent.

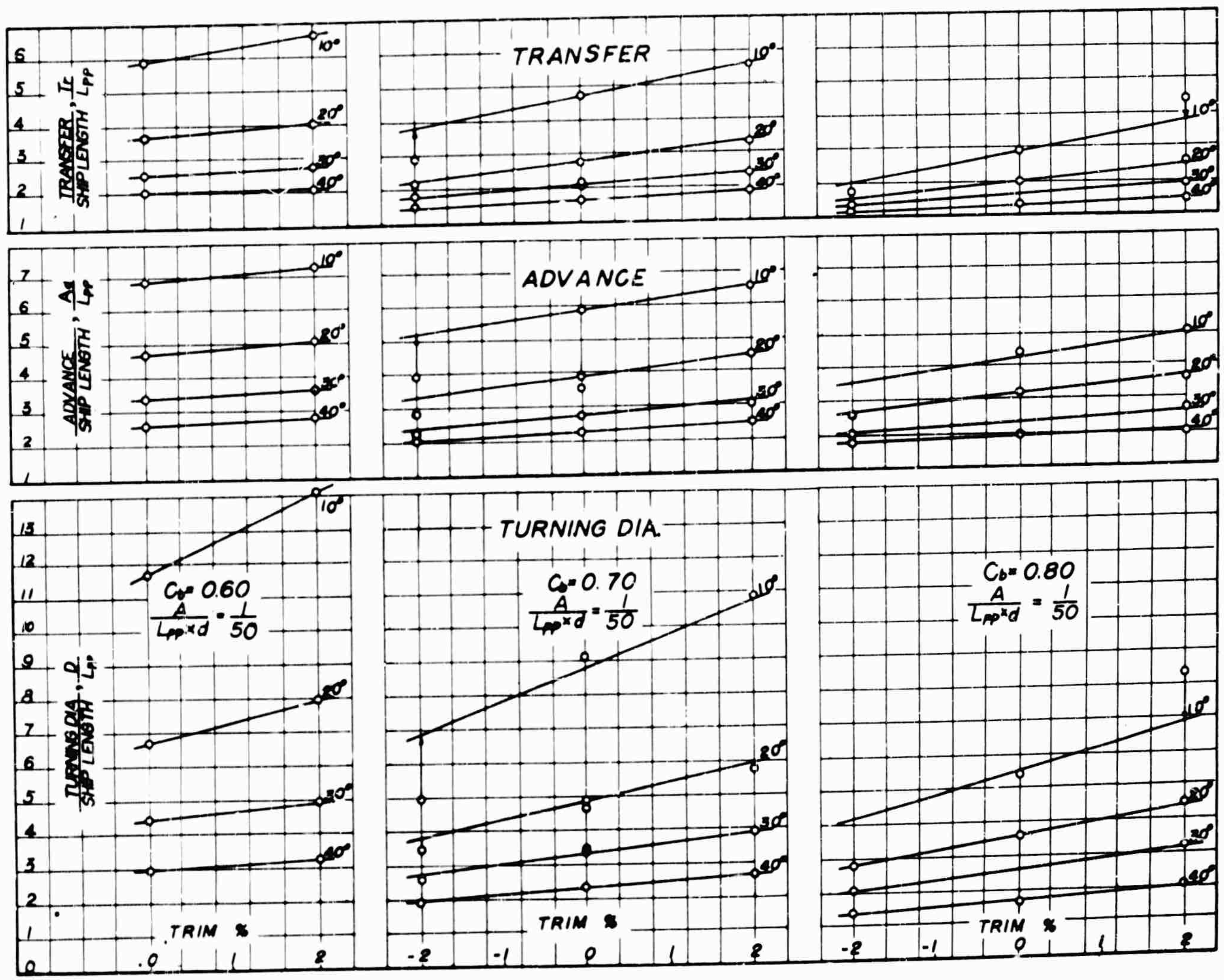


Figure 10 – Effect of Trim upon "Transfer," "Advance," and Turning Diameter

1.7.5 EFFECT OF SPEED

With respect to this problem, we investigated with Models A, B, and C only in the cases when the rudder area ratio is 1/50. The relation between the speed and the turning diameter, obtained by experiments in which the Froude number ranges from 0.09 to 0.30, is shown in Figure 11. Here, the speed while the model is advancing in straight course before steering was taken as the index of the speed. As was expected, the change of the speed affected very little the turning diameter. That is, speed affects very little the turning diameter at the Froude number of less than 0.20. Even in Model C, which has the block coefficient of 0.80, the turning diameter increases by only small percentages when the Froude Number is 0.30. (In another experiment, where we used the model of block coefficient 0.5 in order to make the wavemaking resistance small, the effect of the speed was not recognized until the Froude number became about 0.30.) Relations between speed and advance or transfer are almost the same as that between speed and turning diameter.

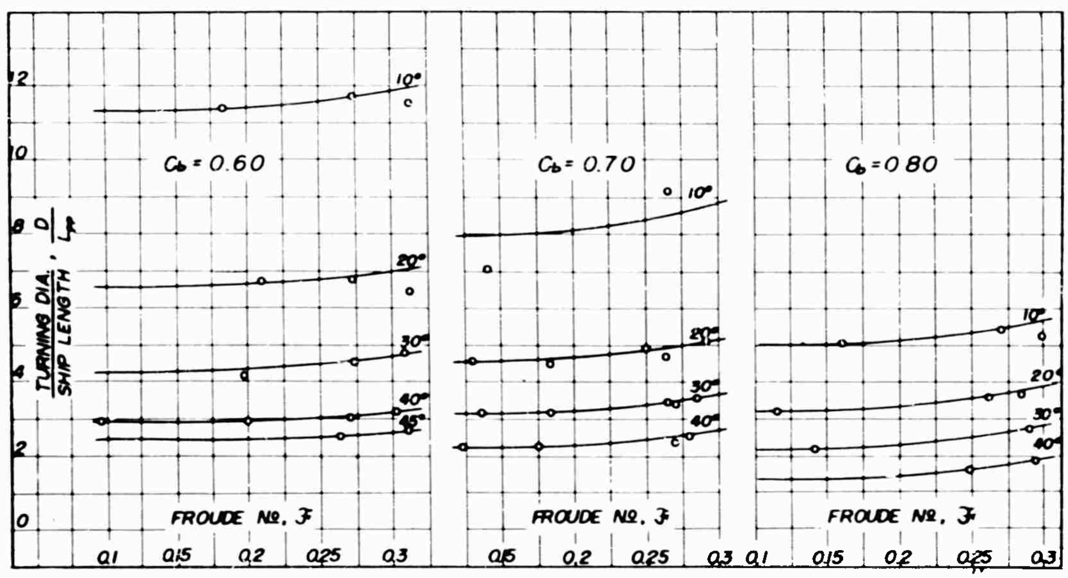


Figure 11 – Effect of Froude Number upon Turning Diameter

1.7.6 EFFECT OF DISPLACEMENT

Figure 12 shows the variation of turning ability due to the change of displacement, keeping their trim unchanged. The displacements correspond to the light- and the full-load condition, respectively, but in order to keep the propeller and the rudder entirely immersed, even in the light condition, the trim by the stern of 1/50 was adopted in both conditions. Accordingly, the displacements and conditions are slightly different from those of the real light- and full-load conditions of an actual ship. From the figure, we can see that even by

increasing the displacement by about 35 percent, the turning diameter changes very little. As these experiments were performed by the same rudder, it must be noticed that the rudder area ratio in the light-load condition is about 30 percent larger than that in the full-load condition.

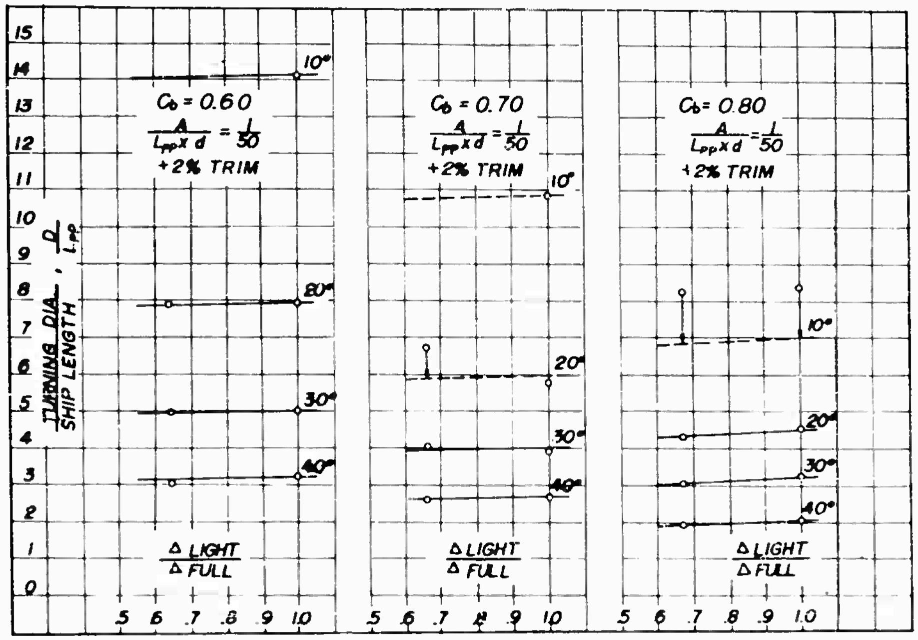


Figure 12 – Effect of Displacement upon Turning Diameter

1.7.7 SPEED REDUCTION IN TURNING

Figure 13-1 shows the speed reduction of the model in the steady-state turning, compared with that in the straight course. The left part of the figure shows the relation between speed reduction and turning diameter, while the right part shows the plots of the relation between speed reduction and rudder angle for each rudder. According to the latter, the speed reduction varies with the rudder angle or the rudder area for the three models. In the former, the relation between the speed reduction and the turning diameter can be represented by a single curve, respectively, for three models in the figure, independent of the rudder angle or the rudder area. Moreover, the tendency of these curves resembles each other in this case. That is, the speed reduction is almost determined by the turning diameter. In Figure 13-2, where the left part of Figure 13-1 is assembled, it is observed that the speed reduction is smaller for the ship with the larger block coefficient.

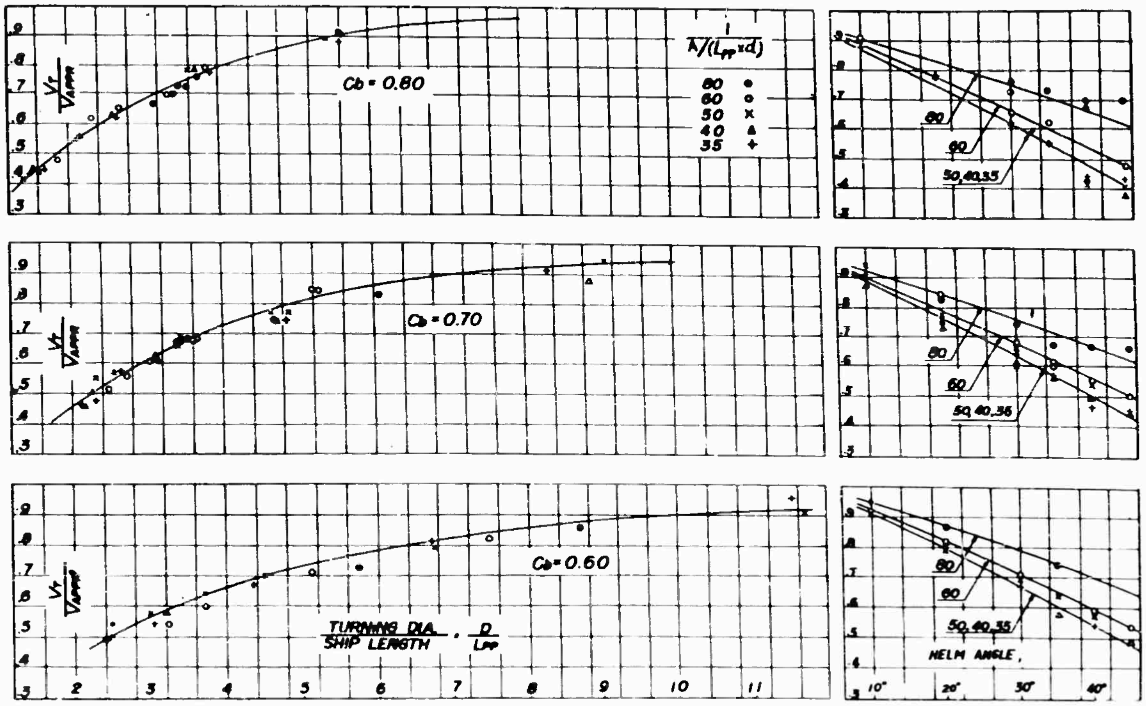


Figure 13-1

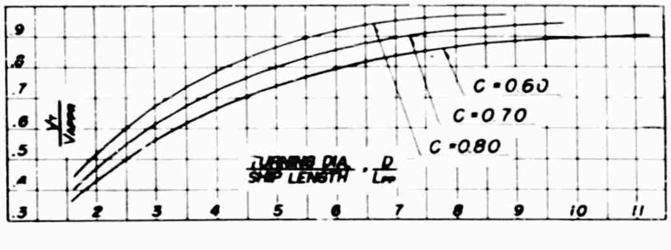


Figure 13-2

Figure 13 – Speed Reduction in Turning

1.7.8 DIFFERENCE OF TURNING PERFORMANCE TO PORT OR STARBOARD

In a single-screw ship, there are some differences in the turning performance between turning to port side and turning to starboard side, due to the nonuniformity of the wake of hull in rudder position and the effect of propeller stream upon the rudder. In this experiment, we investigated the turning to port and compared it with that to starboard, using the propeller of pitch ratio 0.80, and found that the former is about 10 percent smaller in turning diameter than the latter (Figure 14). Since the relative positions of the rudder and the propeller in the models are the same as that of the ordinary ships, as shown in Figure 1, different performance of the

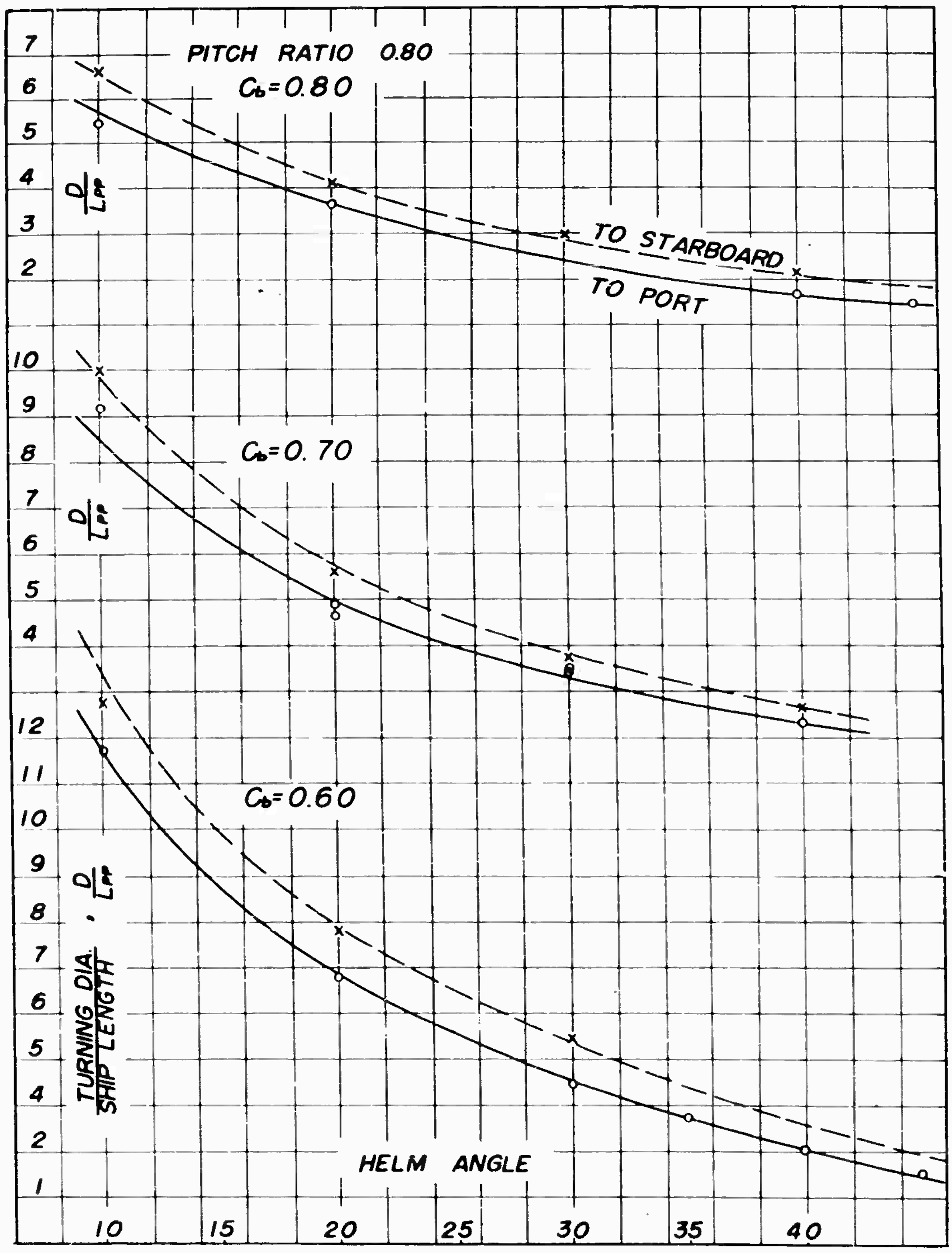


Figure 14 – Effect of Propeller Turning Direction

turning due to the side (port or starboard) can be approximately the same with the results shown in Figure 14, even though we need to investigate it more carefully.

1.7.9 EFFECT OF PROPELLER PITCH RATIO

Three screw propellers of pitch ratio 0.70, 0.80, and 0.90, were used in these experiments, and the results show very little difference. This is probably because the propeller stream is not affected so much in this range of pitch ratio. (Figures about 0.7 and 0.9 in pitch ratio were omitted.)

1.7.10 DRIFT BY WIND FORCE IN TURNING

We have already described in Section 1.5, Results of Experiments, how to correct the effect of drift by the wind in the turning test of models. In the sea trial of the actual ship, also we obtained a good result by applying this same method. As a custom we have observed the turning path for less than one turn in the ship trial. If we observe this about two circles in turning path and correct the drift, however, we shall be able to obtain more precise data about, for example, the turning diameter.

1.7.11 EFFECT OF BILGE KEELS

Existence of the bilge keels naturally affects the turning performance of the self-propelled model. In the experiment performed by the Ship Performance Division, TTRI in 1956, the turning diameter increased by about 10 percent by taking off the bilge keels. Of course, this effect differs according to the length, width, and position of the bilge keels. It is necessary to carry out further experiments to make clear the effect of bilge keels upon the turning ability, because, as described previously, the bilge keel was not used in this series of experiments.

1.8 CONCLUSION

Based on the fact that the result of the turning experiment by the self-propelled model proves the maneuverability of the actual ship, we can conclude several points as follows, from the results of the turning experiment performed by three kinds of models.

1. So long as the rudder area ratio remains less than about 1/45, the maneuverability of the ship is improved by increasing the rudder area. When the rudder area becomes larger than this, however, maneuverability is not improved any more by further increase of the rudder area.
2. In accordance with the increase of the helm angle, the turning diameter, advance, and transfer reduce. This tendency is kept until the large helm angle of around 45 deg. Therefore, in some cases, to increase the maximum helm angle is much more effective to improve the maneuverability than to increase the rudder area only. This tendency becomes very small at the helm angle larger than 45 deg.

3. The hull form affects maneuverability considerably; for example, the turning diameter of the ship of $C_b = 0.60$ is as large as twice that of the ship of $C_b = 0.80$. The tendency that in accordance with the increase of the block coefficient the maneuverability is improved, shows that the problem of turning is closely related to the problem of the course stability. Especially in the case of designing the rudder for full ships, which are usually rather poor in course stability, the character of course stability should be considered simultaneously.

4. The trim of the hull affects considerably the maneuverability. When trim by stern increases by 1 percent, for example, the turning diameter increases by about 10 percent.

5. The change of displacement has very little effect on the turning diameter, if the trim stays the same.

6. The change of the advance speed, if that was in the range of the speed of the ordinary merchant ship, has very little effect on the turning diameter.

7. The speed reduction by the turning can be roughly expressed by a simple curve.

8. The method to measure the steady-turning circle twice is very helpful to correct the effect of drift by the wind in the ship trial, as well as in the model tests.

PART 2

EFFECTS OF SCREW·PROPELLER ON THE CHARACTER OF THE RUDDER

2.1 INTRODUCTION

Even in the design of single rudder for the single-screw ship, where the rudder works behind the screw propeller, because of the lack of the modern design data, we are still using the conventional method of the old-fashioned empirical formula for the open rudder; that is, for the rudder independently working in widely open still water, modifying by our own "experience."

At present, however, with the advanced hull and rudder forms and increased speed, this simple method occasionally gives us erroneous results, if it is not supplemented cleverly by experienced designers.

The author set the target of this paper principally to get practical design data for the rudder which works behind the screw propeller and is under the effect of its aftstream.

Another fact is that the maneuverability of the ship becomes worse when we increase the rudder area too much. This might be acceptable from the theoretical study, but we have no reliable design data to get the best rudder area quantitatively, as well as qualitatively. Accordingly, the author intended also to find out whether the existence of the optimum rudder area depends upon the characteristics of the rudder itself or not.

2.2 MODEL RUDDERS AND PROPELLERS

The identical rudders and propellers, used in the experiments reported in Part 1, were used here, although only one kind of propellers of pitch ratio 0.8 was used.

2.3 METHOD OF EXPERIMENT

In order to avoid the free surface effects, the upper edge of the rudder was kept 10 cm deep under the free-water surface. The characteristics of the rudder were measured by the rudder dynamometer, at the slip ratio of 11 percent, (22 percent), 28 percent, and 44 percent changing the revolution of the screw, at the same advance speed of 1.17 m/sec.

Besides these, the same measurement was undertaken at slip 100 percent. The Reynolds number in this experiment is about 1.0×10^5.

2.4 RESULT OF EXPERIMENT

The normal pressure F and the distance between the leading edge and the center of pressure, are shown in Figures 15 and 16 in the form of C_N (the normal pressure coefficient), and x (the ratio of the distance to the chord length), respectively.

(Text continued on page 83.)

Figure 15 — Normal Pressure Coefficient

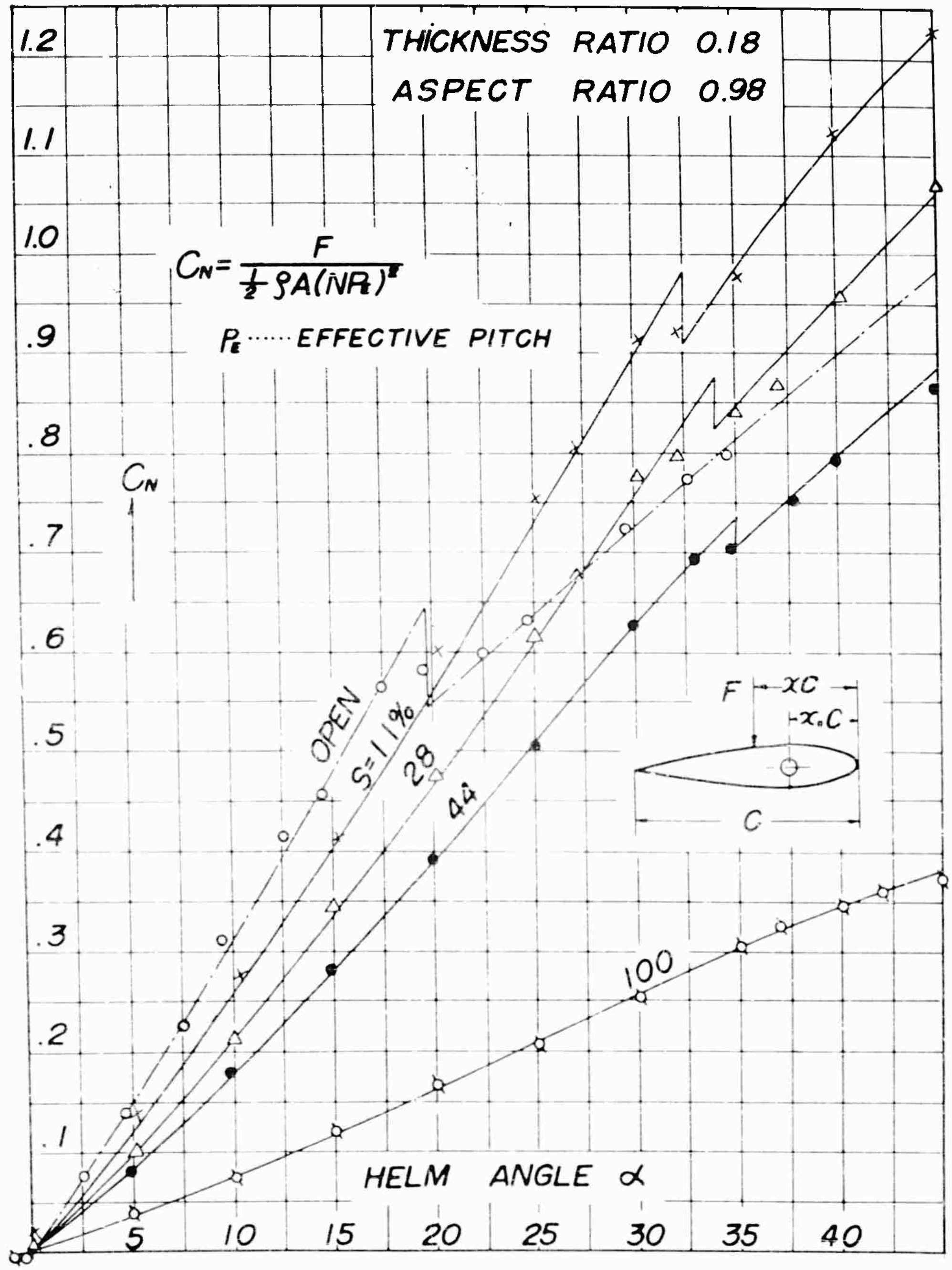
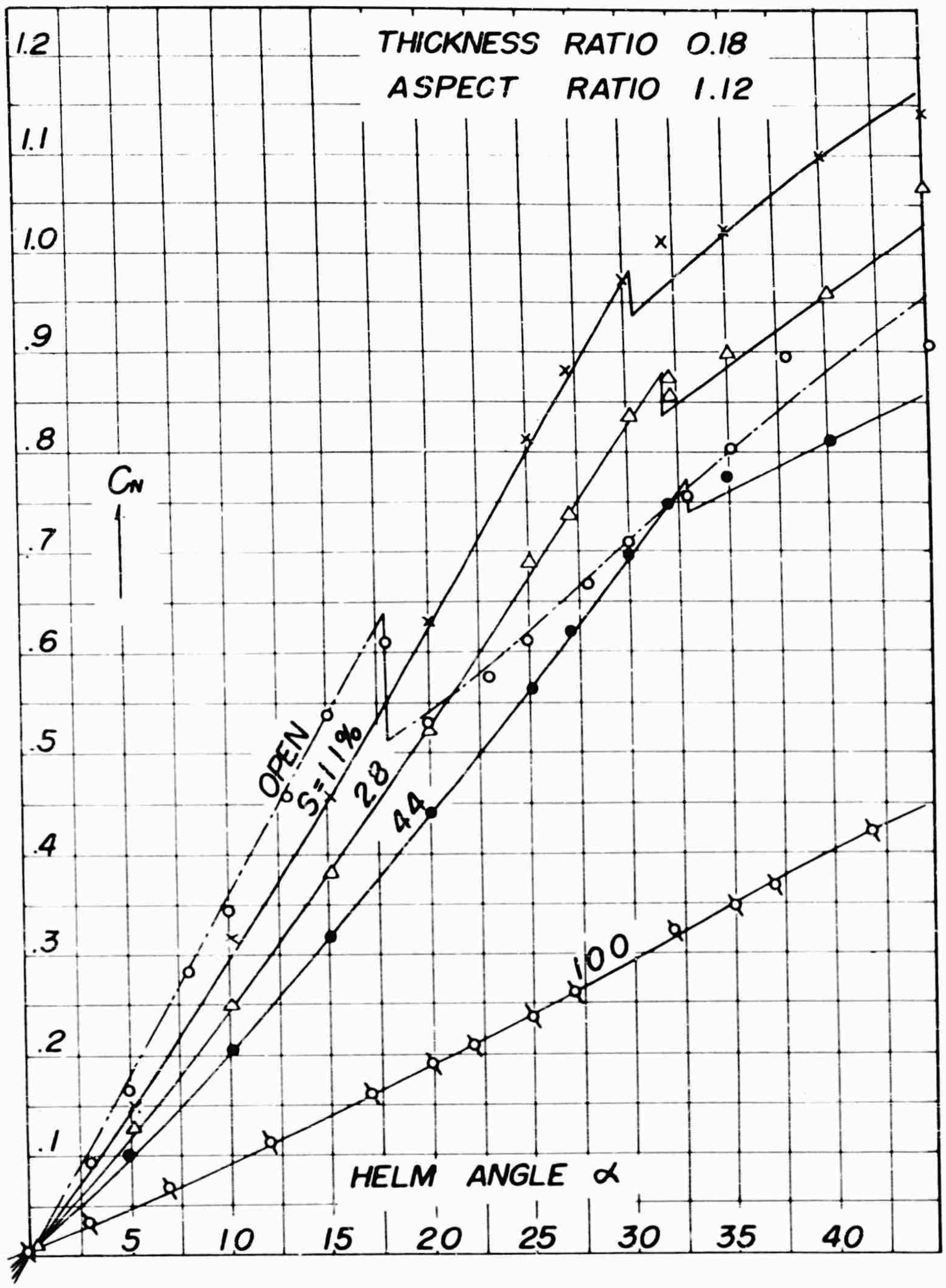


Figure 15-1



THICKNESS RATIO 0.18
ASPECT RATIO 1.12
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
C_N
OPEN
S=11%
28
44
100
HELM ANGLE α
5
10
15
20
25
30
35
40

Figure 15-2

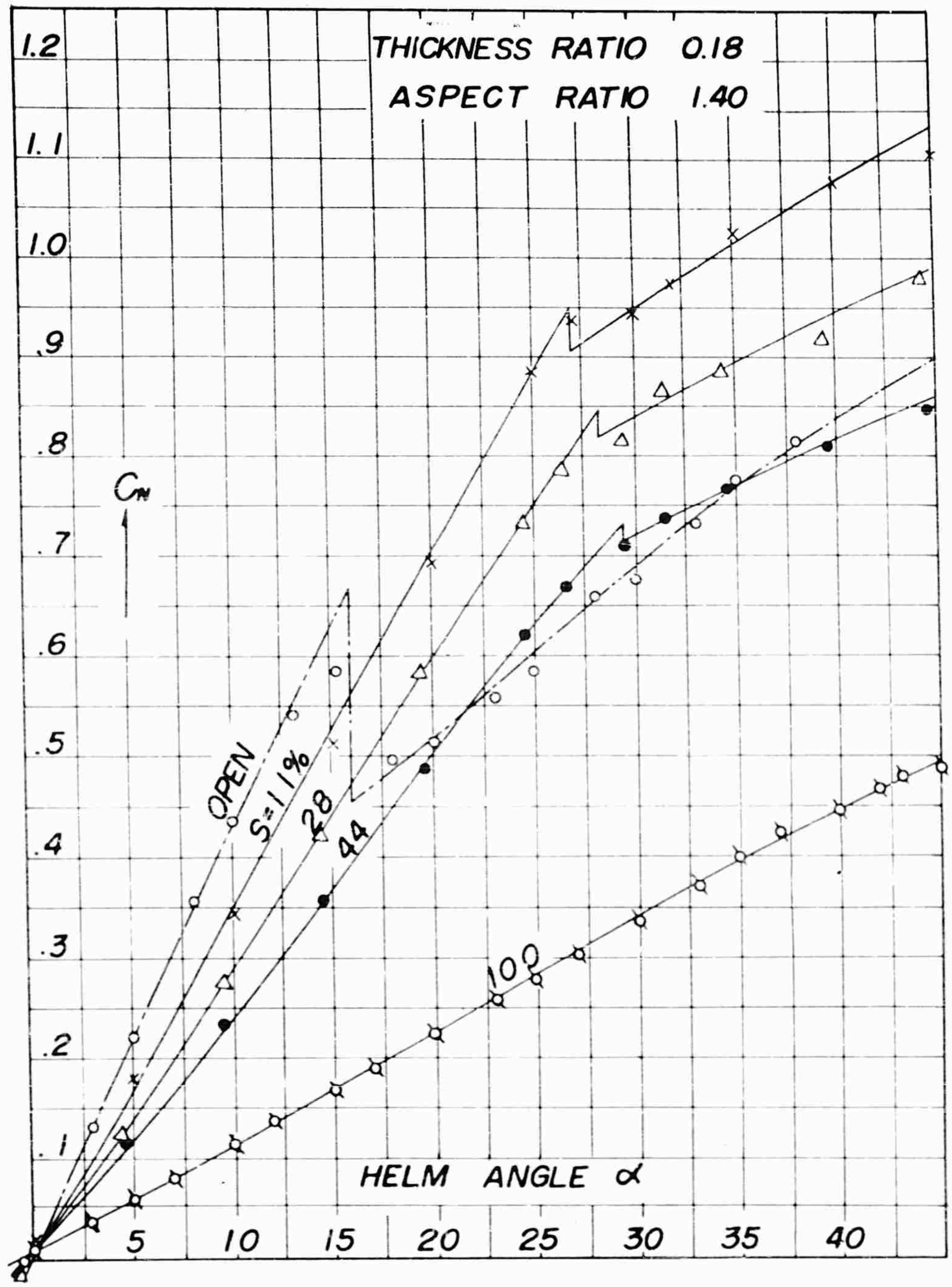

THICKNESS RATIO 0.18
ASPECT RATIO 1.40
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
C_N
OPEN
S=11%
28
44
100
HELM ANGLE α
5
10
15
20
25
30
35
40

Figure 15-3

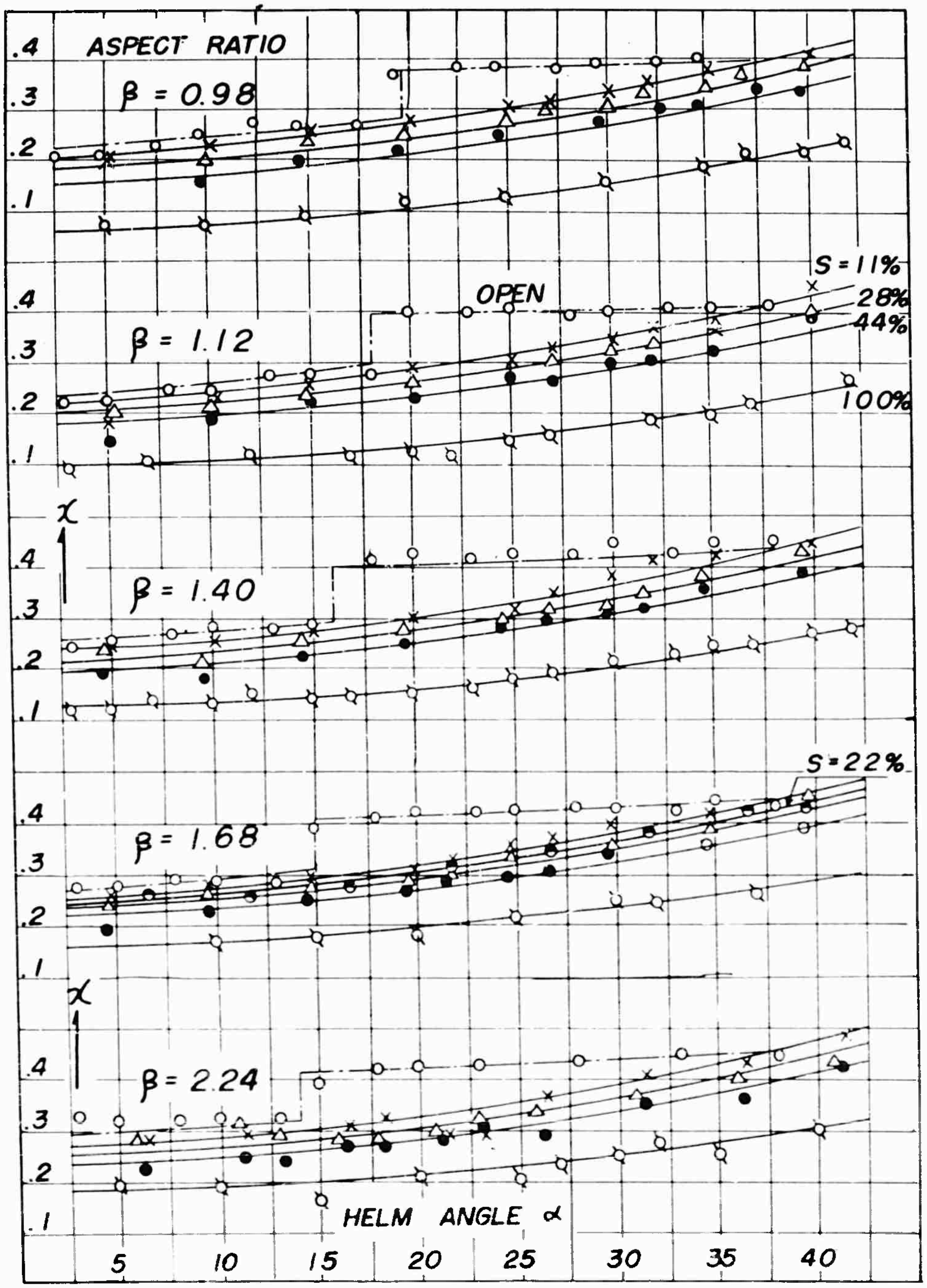


Figure 16 – Center of Pressure

P_E is the effective pitch, and NP_E is adapted as the speed, because the uniform velocity to the rudder can be approximated by the following formula as a whole.

$$V + s\,NP_E = V + (NP_E - V) = NP_E$$

This is also applicable to the case of 100-percent slip ratio.

2.5 REMARKS ON RESULTS

2.5.1 DESIGN DIAGRAMS

For the sake of convenience, the relation between the normal pressure coefficient C_N and the aspect ratio β, is shown in Figure 17-1, the relation between the aspect ratio β and the ratio x of the distance of the center of the pressure from the leading edge to the chord length is shown in Figure 17-2 (or Figure 16).

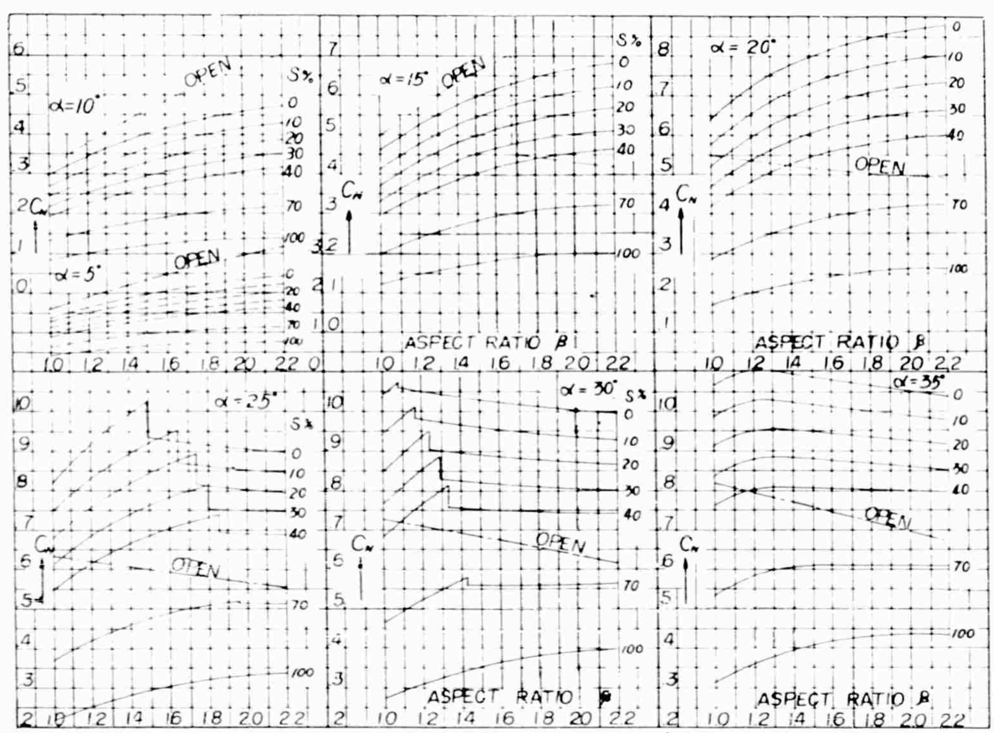


Figure 17-1 – Design Diagrams for Normal Pressure Coefficient C_N

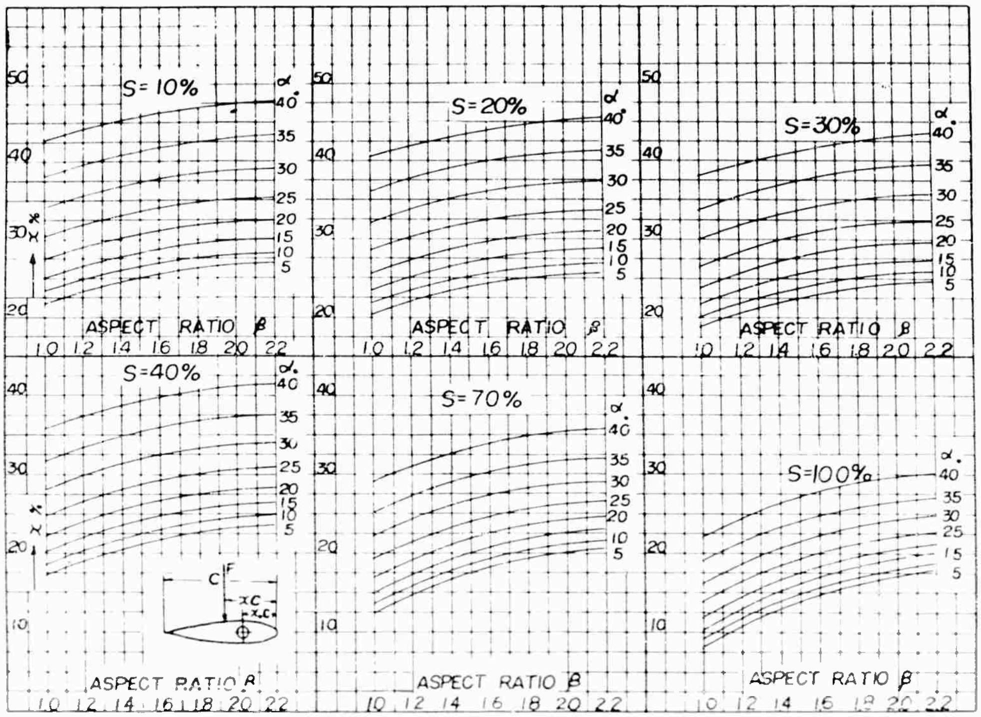


Figure 17-2 – Design Diagrams for Center of Pressure X

2.5.2 EFFECT OF SCREW PROPELLERS ON CHARACTERISTICS OF RUDDER

5.2.1 Normal Pressure Coefficient C_N

Figure 17-1 shows the following facts:

a. As the propeller slip ratio s increases, apparent C_N reduces remarkably.

b. Even in the case of slip ratio 0, in other words, in the lack of aftstream by the screw, C_N reduces remarkably compared with the case of open rudder. This can be regarded as the effect of disturbance by the screw.

c. The abrupt decrease of C_N by stall is remarkably small compared with the case of open rudder.

5.2.2 Change of Center of Pressure

We must note that the position of the center of pressure shifts forward according to the increase of the slip ratio s, as shown in Figure 17-2. In the case of the open rudder, the position of the center of pressure moves immediately aft when the stall takes place.
On the contrary, in case of the rudder which works behind the screw, no abrupt change of this kind can be recognized, at least for the cases of steering to port.

5.2.3 Normal Pressure Coefficient after the Stall

As shown in Figure 16 or Figure 17-1, the normal pressure coefficient of the open rudder decreases considerably by the stall; on the contrary, that of the rudder which works behind the screw does not decrease so much by the stall. It was clearly proved also by the photographic observation that these phenomena were not due to the air draw.

5.2.4 Stall Angle

Figure 18 shows how the stall angle varies with the difference of the propeller slip ratio and the aspect ratio of the rudder. Namely, the stall angle is a function of s and β.

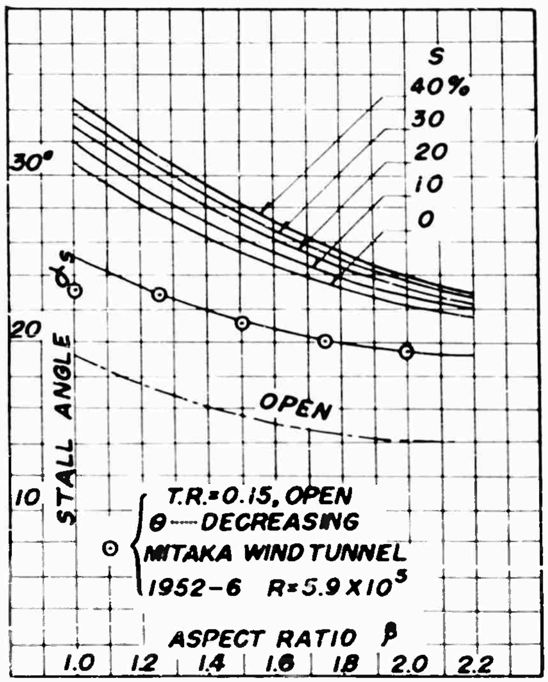


Figure 18 – Stall Angle

As a reference, the experiment result of the airfoil of 15 percent in thickness ratio in wind tunnel is also shown in the same figure. Comparing this value with that of the open rudder, which we got in this series of experiments, difference in stall angle is about 5 deg. However, we are not sure about this difference, as there are differences in fluid, thickness ratio of the foil, method of support, condition of the vibration, fineness of finishing of the surface, existence of the free surface, and method of experiments. It is, however, assured also by another experiment that stall angle changes considerably by the effect of the screw.

5.2.5 Considerations on Rudder Torque

As mentioned above, C_N and x are clearly proved to be a function of s and β. Therefore, it is not enough to design the rudder using the simple empirical formula, which does not include the effect of s and β. If we assume the position of the rudder head as usual, and calculate the rudder torque from Figure 17, at various propeller slip ratios, we can find easily that the rudder torque can be in overbalance condition at large slip ratio s. This means, in case of the design of rudders for tugboats and trawlers, we have to be careful in the decision of the position of the shaft of the rudder.

2.6 CONSIDERATION FOR BEST RUDDER AREA

The expression of the normal force F can be manipulated as follows:

$$F = \frac{1}{2}\rho A V^2 C_N = \frac{1}{2}\rho(2b)^2 V^2 \frac{C_N}{\beta}$$

As is clear from this expression, F changes by the values of C_N/β, if $2b$ remains constant (because Span $2b$ of the rudder is usually limited to some value which comes from the form of the stern). That is, to increase the area under this limitation means to reduce the value of β, and this results to decrease the value of C_N. Accordingly, we cannot say, in general, how the value C_N/β changes by the increase of the rudder area. Figure 19 shows the lift coefficient of the airfoil Clark Y with the thickness of 11 percent which was acquired by Zimmerman

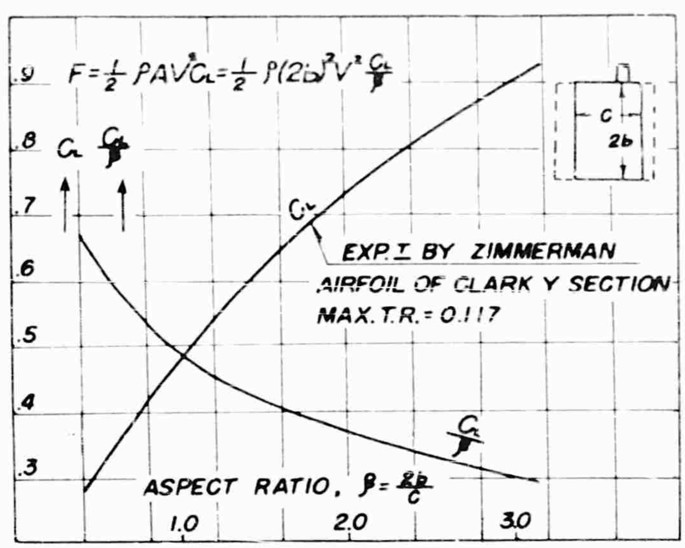


Figure 19 – Effect of Rudder Area upon Lift

in the wind tunnel. In the same figure, the value of C_L/β is also shown. As C_N is almost the same with C_L, C_L was used in this figure. From this figure, we can find the tendency

that C_L/β, namely, the total lift increases as β reduces (the area increases), notwithstanding the fact that C_L decreases at the same time.

C_N changes by the difference in airfoil section, wing thickness ratio, and by the existence of the screw. In Figure 20, the value of C_N/β was calculated from the result of this experiment in which a symmetrical airfoil of 18 percent in thickness was used.

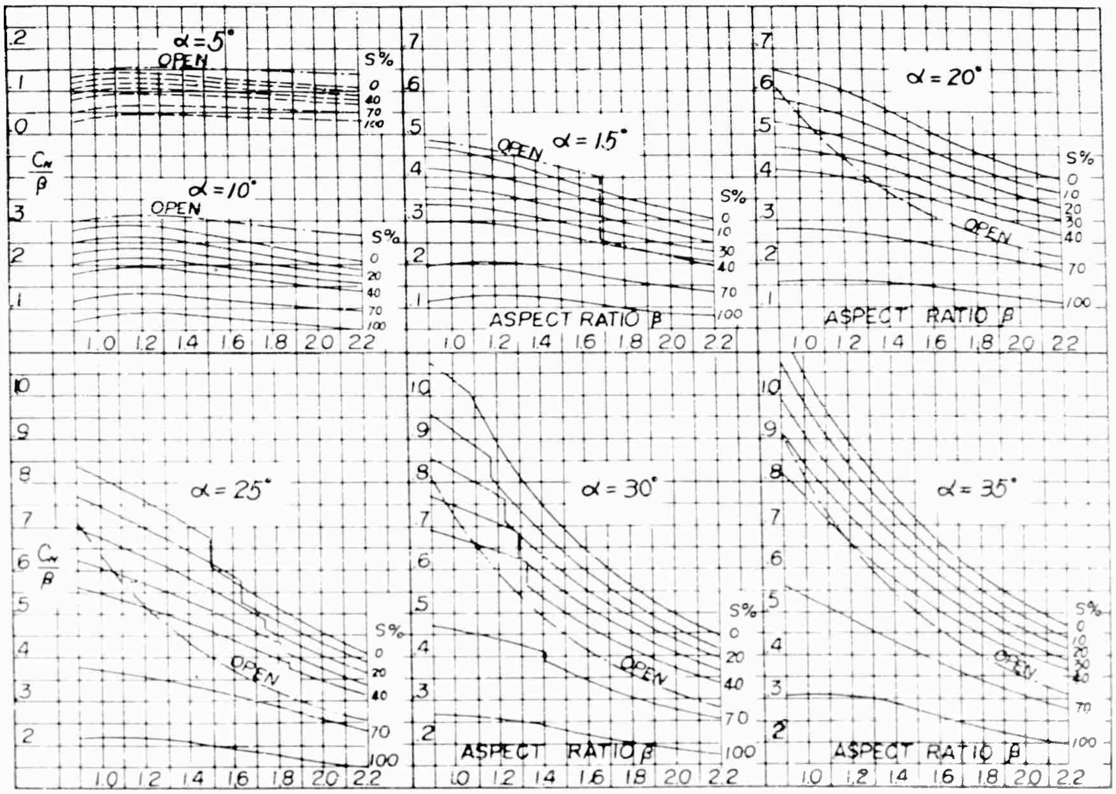


Figure 20 – Curves of C_N/β

(Rudder thickness ratio of the ordinary rudder is about 18 percent.) From this, we can see that at the helm angle less than 15 deg, C_N/β increases by the decrease of β until β comes to some value; however after that, this tendency stops and C_N/β begins to fall down or in other words C_N/β takes its maximum value at some value of β. On the other hand, when the helm angle is more than 15 deg, C_N/β always increases by the decrease of β, and this tendency is especially clear after stall. This is also the same in the results of the open rudder tests. Now, let us consider the best rudder area ratio again which we discussed in Part 1. Figure 21 shows the steady turning diameter ratio at the helm angle of 35 deg. From this, we can say that if the existence of the optimum rudder area ratio comes only from the characteristics of the rudder, the true angle of incidence to the rudder must be less than 15 deg, even

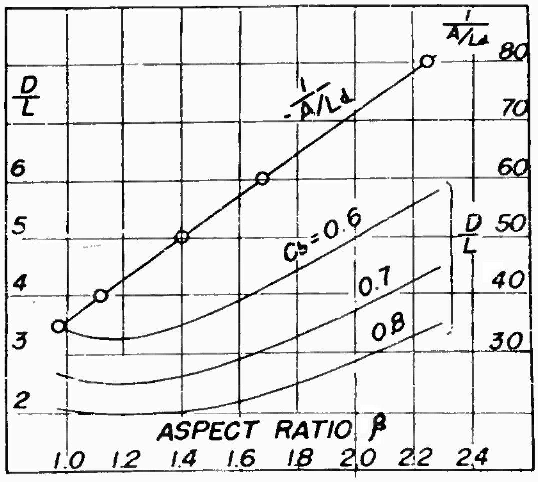


Figure 21 – Turning Diameter at 35 Degrees of Helm Angle

at the turning by helm angle 35 deg, after the turning becomes steady. The measured values of the drift angle in the previous series of steady turning tests are shown in Figure 22-1. From this figure, we can get the drift angle of the ship when the rudder of rudder area ratios 1/50 and 1/60 were used (Figure 22-2). Using these results, we can get, while the ship is turning steadily by the helm angle of 35 deg, the true angle of incidence of the rudder of helm angle 35 deg for the ship of block coefficient 0.7 and 0.8, and rudder area ratio of 1/50 and 1/60, respectively, as are shown in Table 5. That is, we can see that, even in the steady turning by the helm angle of 35 deg, the angle of incidence to the rudder is less than 15 deg. Considering the facts shown in Figure 20, we can say that the best rudder area depends on the characteristics of the rudder itself.

TABLE 5

Block Coefficient	C_b	0.7		0.8		
Length of Model (m)	L	2.5		2.5		
Helm Angle (degree)	α	35		35		
Rudder Area Ratio	$\frac{A}{L \times d} = a$	1/50	1/50	1/50	1/50	
Aspect Ratio	β	1.40	1.68	1.40	1.68	
Radius of Turning Circle (m)	ρ_0	3.37	3.88	2.50	2.88	Figure 21
Drifting Angle at C.G. (degree)	θ_0	14.5	12.5	19.5	17.4	Figure 22-2
Drifting Angle at Rudder Position (degree)	θ_1	32.8	29.0	41.5	37.5	
Real Slip Ratio of Propeller	s	0.35		0.40		
Correction Angle Due to S (degree)	θ_2	7.4	6.7	9.2	8.8	
Incidence Angle of Flow to Rudder (degree)	θ_3	9.6	12.7	2.7	6.3	$= \alpha - (\theta_1 - \theta_2)$

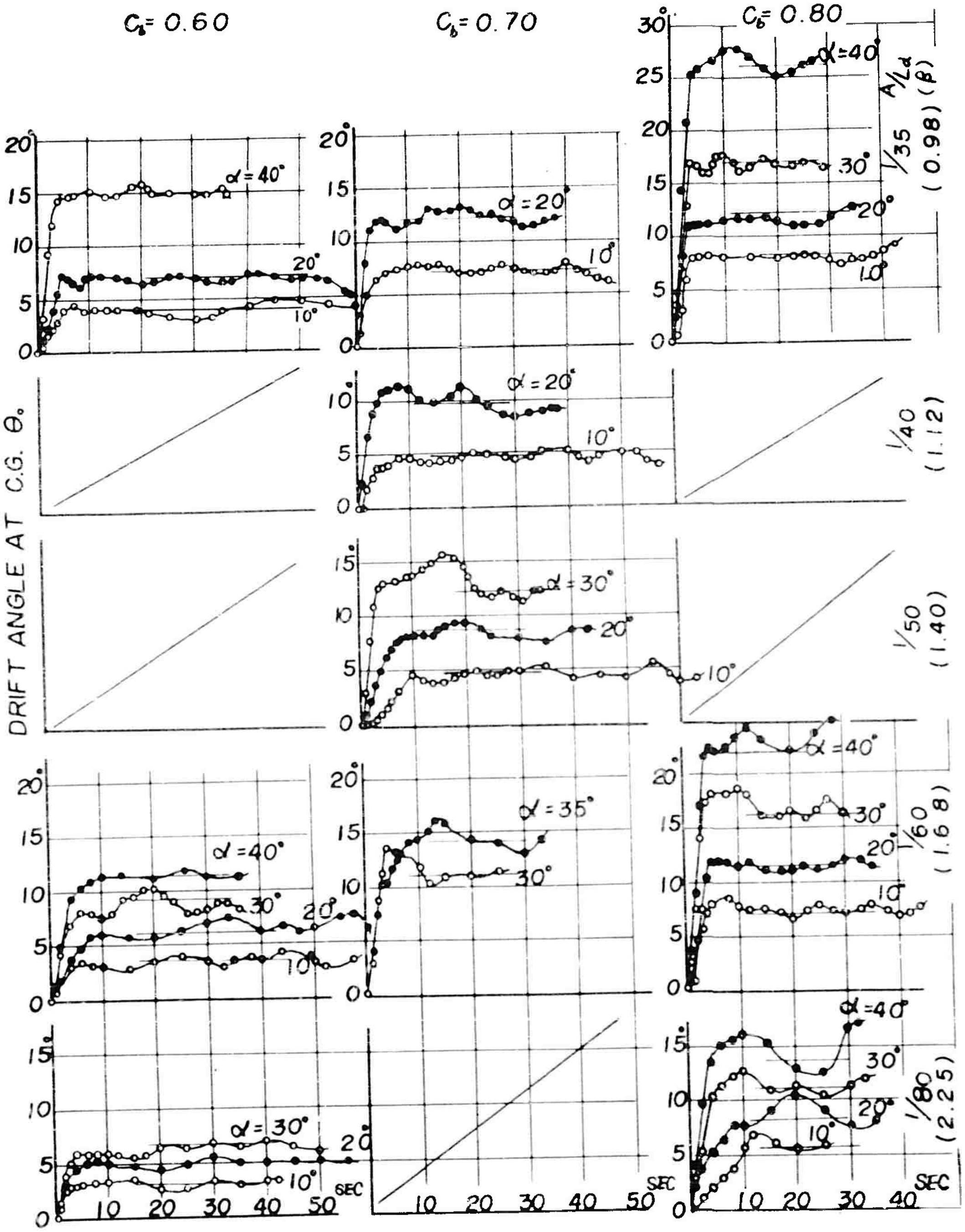


Figure 22-1 – Drift Angle Measured

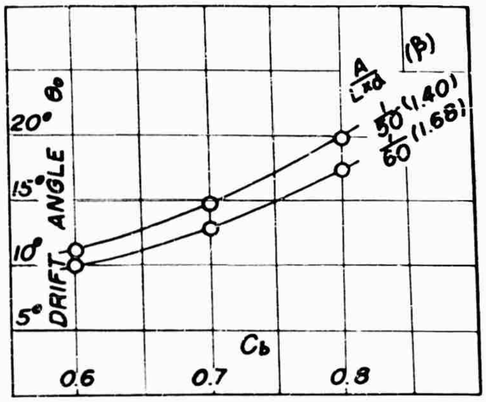


Figure 22-2 – Drift Angle at 35 Degrees of Helm Angle

2.7 CONSIDERATION OF THE CONVENTIONAL METHOD OF RUDDER DESIGN

For example, in order to decide the diameter of the rudder shaft, we have to know first the normal pressure F and the distance of the center of the pressure from the leading edge of the rudder. There are many formulas which give these values. Most of them are, however, represented by the following formula:

$$\left.\begin{array}{ll} F = 58.8\, AV^2 \times \sin\alpha & \text{equation by Beaufoy} \\ xC = (0.195 + 0.305 \sin\alpha)C & \text{equation by Joessel} \end{array}\right\} \text{in Kg. M.S. unit}$$

where V is the uniform velocity to the rudder,

A is the rudder area,

C is the chord length of the rudder, and

α is the rudder angle.

It is very convenient in this formula to relate V to the ship's speed V_s. That is, substituting $V = (1 + k)\, V_s$, k is an empirical factor, and expressing the normal pressure and the rudder torque by the form of coefficients, we will get the following formula:

$$C_N' \equiv \frac{F}{AV_s^2} = 58.8\,(1 + k)^2 \sin\alpha$$

$$C_M' \equiv \frac{M}{CAV_s^2} = C_N' \times (x - x_0)$$

where x_0 is the ratio of the distance of the center of the rudder stock from the leading edge to the chord length C of the rudder. The empirical factor k differs according to the different shipyards which design the rudder, and was found to range between 0.05 and 0.15 by the investigation of the Rudder Committee of the Marine Association. In this empirical formula, we cannot take account of the effects of C_b, w, s, and β separately. All of these effects are

represented by only one empirical factor k. Also as for the position of the center of pressure, the effects of the slip ratio s and β are entirely disregarded. C_N' and C_M' by these expressions are shown in Figure 23 by broken lines.

Now, let us compare the result of our experiment with these values. Substituting the relations $1 - s = V_{AD}/NP_E$, and $1 - w = V_{AD}/V_s$ in the expression $C_N = F/\frac{1}{2}\rho A\ \overline{NP}_E^2$ which is used in Figures 15 and 17 to express the normal pressure coefficient from the result of our experiment, we will get the relations:

$$C_N' \equiv \frac{F}{AV_s^2} = \frac{1}{2}\rho\ C_N\ (\alpha,\ \beta,\ s)\left(\frac{1-w}{1-s}\right)^2$$

$$C_M' \equiv \frac{M}{CAV_s^2} = C_N'(x - x_0)$$

In order to compare the experiment results with the calculated value by the conventional formula, we have to know first the value of w and s, and then calculate the value of C_N' and C_M', getting the value of C_N from Figure 17-1, corresponding to the slip ratio s and the aspect ratio β. The value of $(1 - w/1 - s)^2$ depends mainly on the ship form, and we will calculate these values for the general cargo ship of $C_b = 0.7$ and oil tanker of $C_b = 0.8$. See Table 6. In general, both w and s increase as the ship form becomes fuller, and so the value of $(1 - w/1 - s)^2$ does not change remarkably as a result. As is guessed from Table 6, we can even assume this value to be constant 1.27 without serious mistake. Using this value 1.27 for $(1 - w/1 - s)^2$, we calculated the values of C_N' and C_M' for two cases of $C_b \times s = 0.7 \times 0.35$, and 0.8×0.40 when β is 1.80, and showed them by solid lines in Figure 23. Here, x_0 is taken as 0.295 always and the value of x is shown in Figure 23. From this figure, we can see that C_N' is remarkably underestimated, even if k in the formula is taken as 0.15. As the center of pressure from the formula is also quite different from that of the experiment result, the rudder torque coefficient C_M' is much smaller than the actual value for the case of large helm angle as well as for the case of overbalance condition. When this formula is used, we must note this fact: Taking the case of general cargo ship ($s = 0.35$, $\beta = 1.8$) and helm angle of 35 deg as an example, if we want to get the value C_M' from this empirical formula and make this value agree well with the result of the experiment, we have to take the value of k as large as 0.30. This fact shows us more clearly that we have to be very careful in the usage of the conventional formula in rudder design. Of course, when the rudder is steered to a certain angle, the ship will have some drift angle which will act to reduce the incidence angle to some extent, and make the torque a little less than the value given by Figure 23. In the design, however, we had better neglect this reduction.

TABLE 6

C_b	w	s	$(1 - w/1 - s)^2$
0.7	0.27	0.35	1.27
0.8	0.32	0.40	1.27

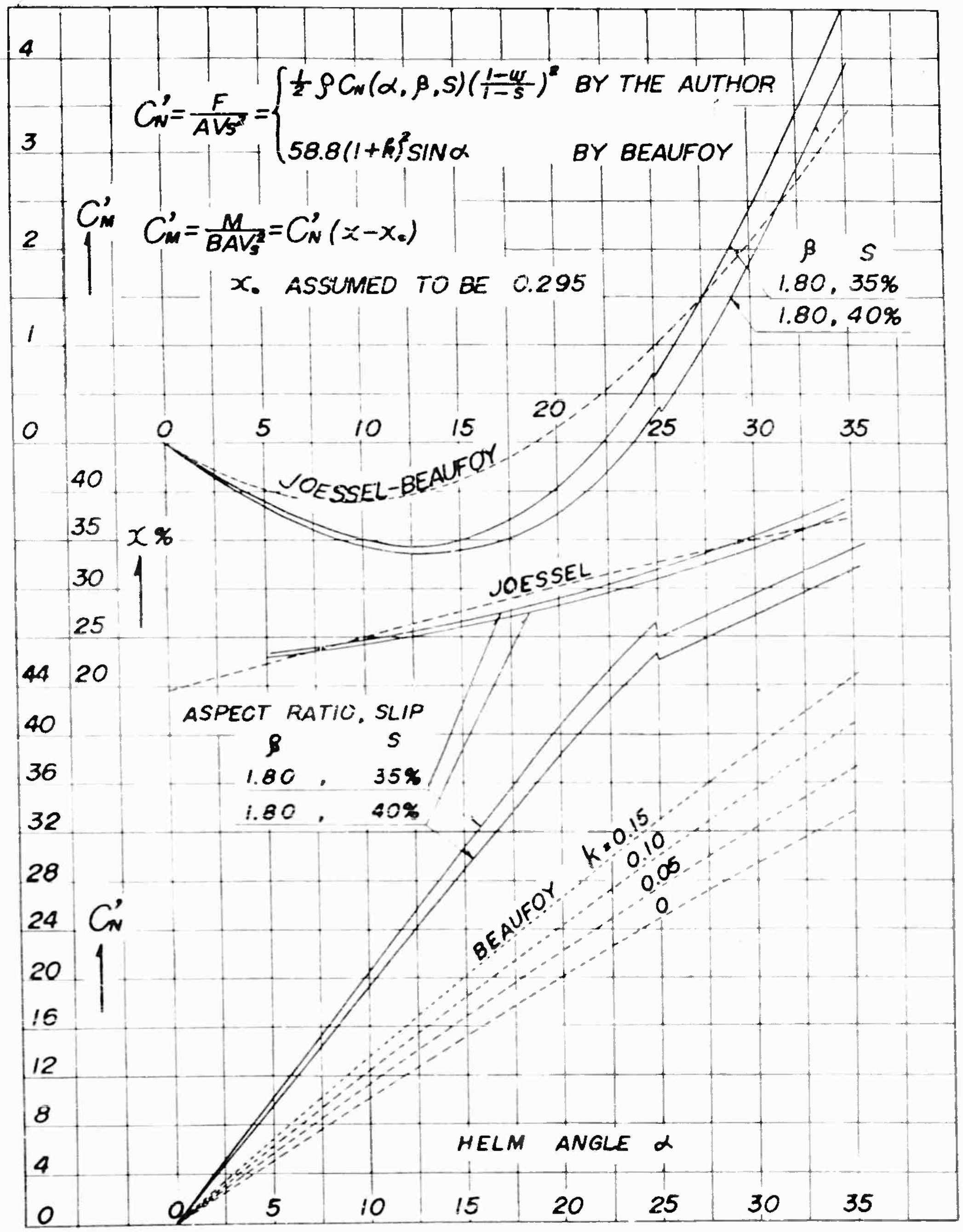


Figure 23 – Comparison with the Calculated Values by Conventional Formula

2.8 CONSIDERATION OF RUDDER TORQUE RECORDED ON SEA TRIAL

Now we like to investigate the applicability of our design data to the actual case of ship rudder. The left and the right columns in Figure 24 show the time history of the rudder torque, measured on each 5 ships by the magneto-striction type torsion meter, by the Ship Performance Division and Hidachi Technics Research Institute, respectively. As these are all the general type of cargo ships, we assume that they can be represented by one ship as follows, for simplicity:

Block coefficient	0.7
Wake factor	27 percent
Drift angle at the position of the rudder	3 deg
Rudder thickness ratio	18 percent
Slip ratio, s	35 percent
Correction of angle for s	1 deg
Density of salt water, ρ	1.025×101.96
Real drift angle	2 deg

(Rudder section is symmetrical airfoil type)

Using these values, Table 7 is obtained from Figure 17. Then the torque M can be calculated by the following formula:

$$M = \frac{1}{2}\rho\left(\frac{1-w}{1-s}\right)^2 C_N\,(x - x_0)\;CAV_s^2 = C_N'\,(x - x_0)\;CAV_s^2$$

The results of these calculations are given in Table 8 and plotted in Figure 24 at the time when steering was just finished. The mark 0 shows the case when the drift angle is assumed to be 3 deg. As found from Figure 24, these values agree comparatively well with the measured values, except for the case of the NICHIYO-MARU. Accordingly, from the practical point of view of designing the rudder, we can consider as follows:

TABLE 7

α	C_N'	x
35 deg	55.1	0.378
33 deg	53.5	0.365

1. In the general merchant ship, for the purpose of evaluation of the rudder torque, we can use the value of C_N and x of deeply immersed rudder, taking the rudder area as it is, disregarding the effect of the depth of immersion.

2. We can disregard the small drift angle to the rudder which takes place right after the steering is finished.

3. If we want to use the formula of Beaufoy-Joessel, 0.30 should be adopted as the value of the empirical factor k.

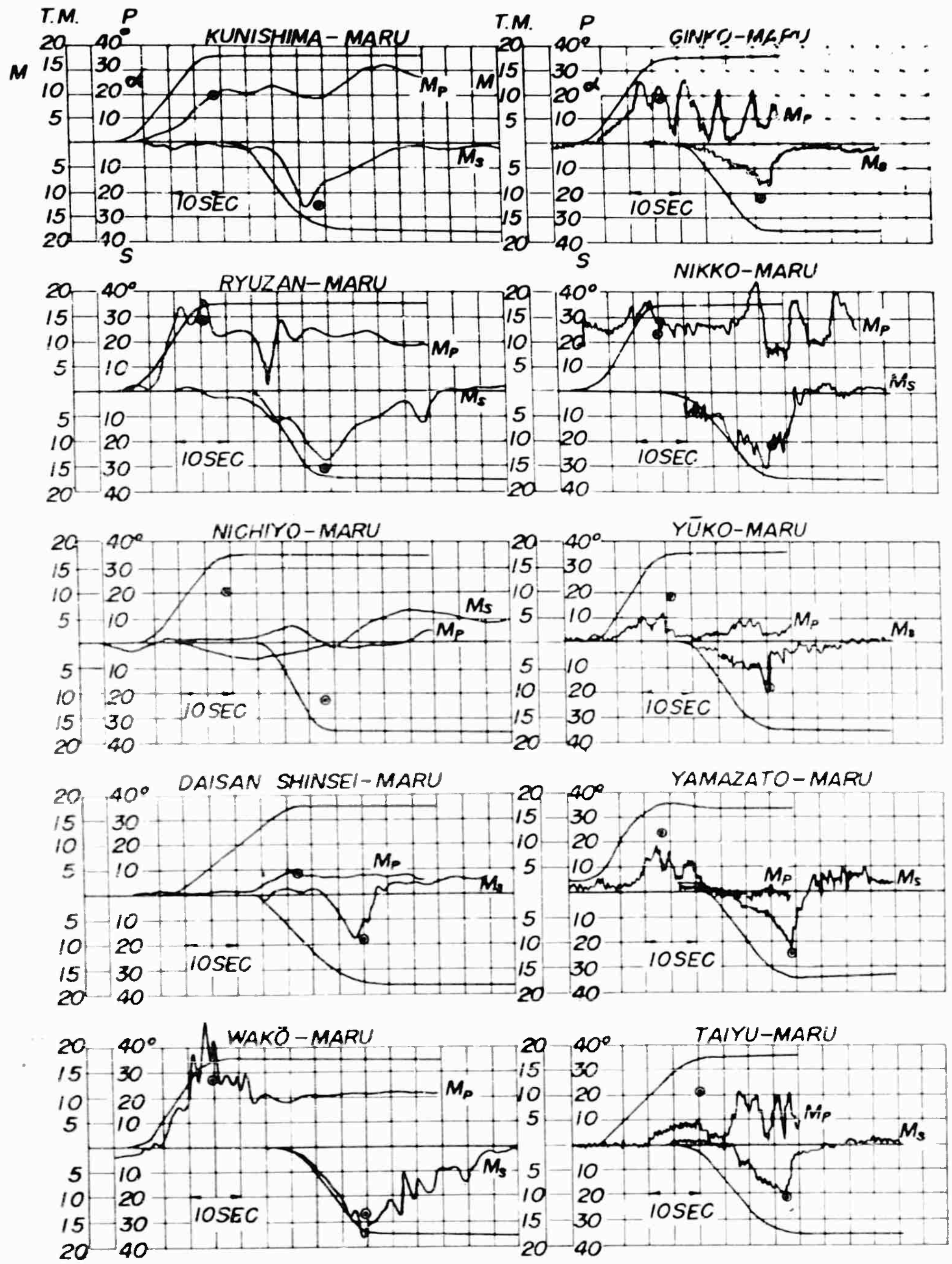


Figure 24 – Twisting Moment of Rudder at Ship Trial

TABLE 8

Ship Name	Chord Length C(m)	Center of Rudder Stock x_0	Rudder Area $A(\mathrm{m})^2$	Ship Speed V_s(m/s)	Twist. Mt. M-Ton Drift Angle 2 deg (real)
KUNISHIMA-MARU	3.05	0.289	16.31	S 7.87 P 6.89	12.5 9.6
RYŪZAN -MARU	3.05	0.289	16.31	S 8.64 P 8.59	15.1 14.9
NICHIYŌ -MARU	2.94	0.297	15.27	S 8.23 P 7.92	11.1 10.2
SHINSEI -MARU	3.05	0.312	16.04	S 7.77 P 5.55	8.4 4.3
WAKŌ -MARU	3.05	0.289	16.31	S 8.28 P 8.28	13.8 13.8
GINKŌ -MARU	2.71	0.273	13.22	S 7.72 P 7.51	10.6 9.9
NIKKŌ -MARU	3.25	0.308	19.01	S 7.30 P 7.72	10.1 11.2
YŪKŌ -MARU	2.66	0.276	12.73	S 7.41 P 7.54	8.9 9.2
YAMAZATO-MARU	3.04	0.282	16.11	S 7.10 P 7.54	11.3 12.8
TAIYŪ -MARU	2.97	0.289	15.91	S 7.20 P 7.61	9.9 11.1

As shown in the torque curves of Figure 24, a sharp hump appears right after the steering finishes. This is very natural because the normal pressure decreases suddenly as the results of the abrupt increase of the drift angle which happens right after the steering is settled. In short, this peak is merely the one which comes from the large value of the incident angle that exists for only a short period of time.

2.9 MISCELLANEOUS

2.9.1 EFFECT OF SCREW PITCH RATIO

It seemed possible to neglect this effect, at least in the range of pitch ratio of 0.7–0.9.

2.9.2 EFFECT OF WAKE DISTRIBUTION AND EFFECT OF DEPTH OF RUDDER IMMERSION

As already shown in the results of the sea trial, these apparently have little effect in the calculation of the rudder torque. The effect of the depth of immersion is treated in Part 3.

2.10 APPLICABILITY OF RESULT OF THIS EXPERIMENT

The screw and rudder, which are used in this experiment, and their relative position correspond to that of the general merchant ship or the super-oil-tanker of full block coefficient. Accordingly, we cannot apply these results from the ship which has extraordinary values in its rudder thickness ratio or number of revolutions of the main shaft, that is, to the ship which has the abnormal relation between the propeller diameter and the rudder dimensions.

2.11. CONCLUSION

The author believes that we made clear two points which we intended to do, as mentioned in the Introduction. That is, he believes we made clear that the best rudder area depends on the characteristics of the rudder itself, and that we could offer the design chart for the rudder, by which it is possible to consider the effect of wake of the ship hull and also the effect of slip ratio of the screw.

PART 3

EFFECTS OF THE DEPTH OF SUBMERGENCE OF THE RUDDER ON ITS PERFORMANCE

3.1 INTRODUCTION

The design chart which we have already offered is for the deeply immersed rudder. The fact that the rudder torque calculated by this chart agrees comparatively well with the rudder torque measured in the actual ship is already shown in Part 2, using some examples. However, as the depth of immersion of the rudder is usually not so large in general cases, it is unreasonable to use this chart to calculate the characteristics of the rudder which is not immersed deeply. The fact that they agreed fairly well, seemed to come from some other reason. Here, we performed again a systematic series of model experiments to find these reasons. This part reports these results.

3.2 MODEL RUDDERS AND PROPELLERS USED

Three model rudders used in this experiment were selected from the five rudders mentioned in Part 1, and the aspect ratios are respectively 1.12, 1.40, and 1.68. The model screw is one of the three model screws also mentioned in Part 1, and the pitch ratio is 0.8.

3.3 SETUP AND RESULT OF EXPERIMENT

The experimental setup is the same as that described in Part 2, but the depth of immersion was varied.

Because the performance of the rudder can be affected by the direction of turning, port or starboard, when it is not sufficiently immersed, we carried out the tests of the turning on both sides in this series of experiments, whereas in the experiments in Part 2, when the rudder was sufficiently immersed, we tested port side only. We showed the results of our measurement in Figure 25 in the form of the normal pressure coefficient.

3.4 CONSIDERATIONS ABOUT RESULT OF EXPERIMENT

3.4.1 RELATIONS BETWEEN NORMAL PRESSURE OF RUDDER AND DEPTH OF IMMERSION

Figure 26 shows the effects of the depth of immersion on the normal pressure coefficient of the rudder, which were obtained from Figure 25. Figure 27 shows the relations between $C_N/C_{M\infty}$ and $l/2b$, in order to help understand the depth of immersion. Symbol ∞ shows the case when the depth of immersion is big enough. That is, as found from Figure 27, except in some special cases the effect of the depth of immersion on the rudder seems to disappear when $l/2b$ becomes larger than about 0.9. As is easily found from Figure 27, also the effect of the depth of immersion increases as the slip ratio of the propeller becomes larger.

(Text continued on page 105.)

Figure 25 – Normal Pressure Coefficient When Rudder Depth Varied

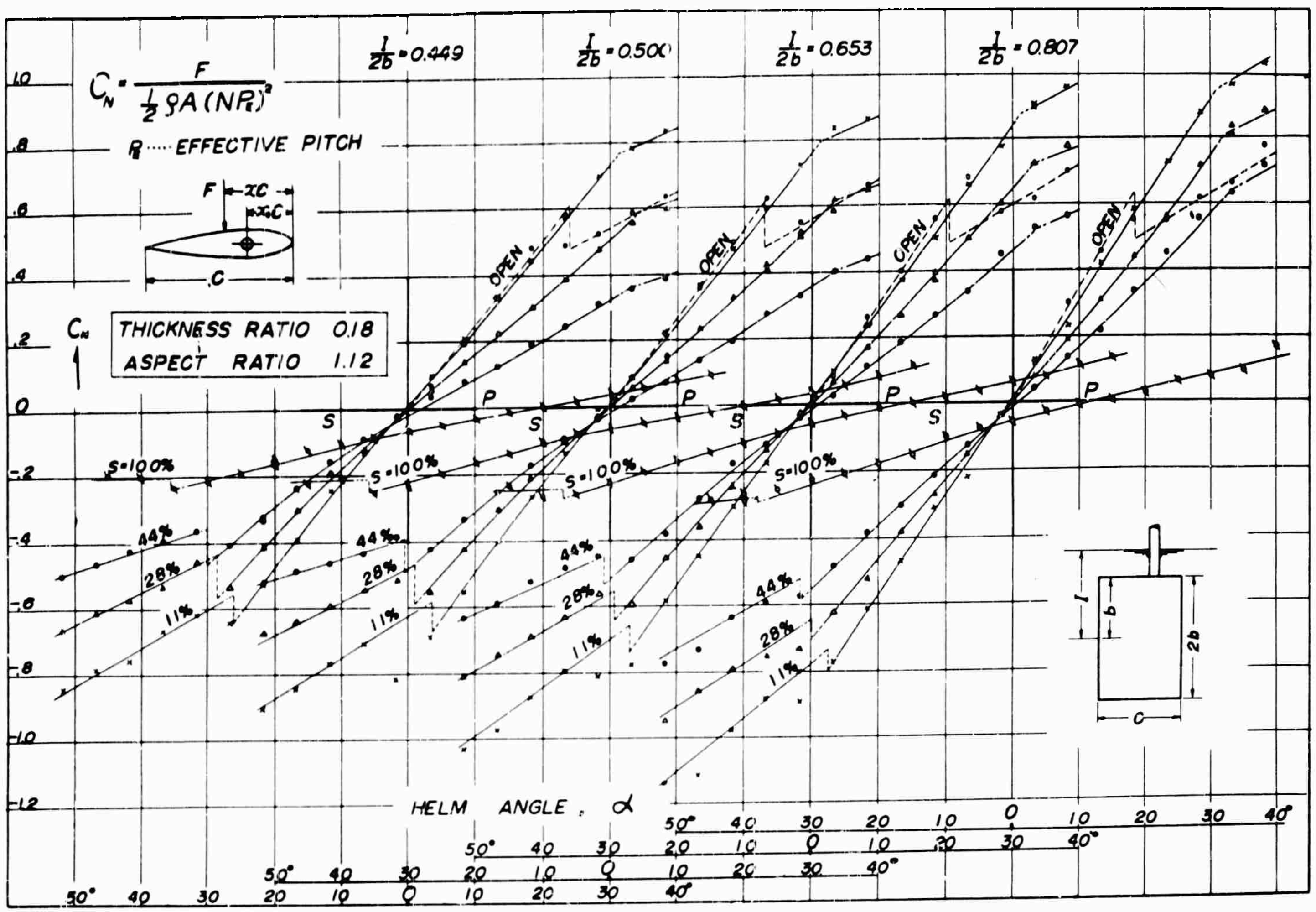


Figure 25-1

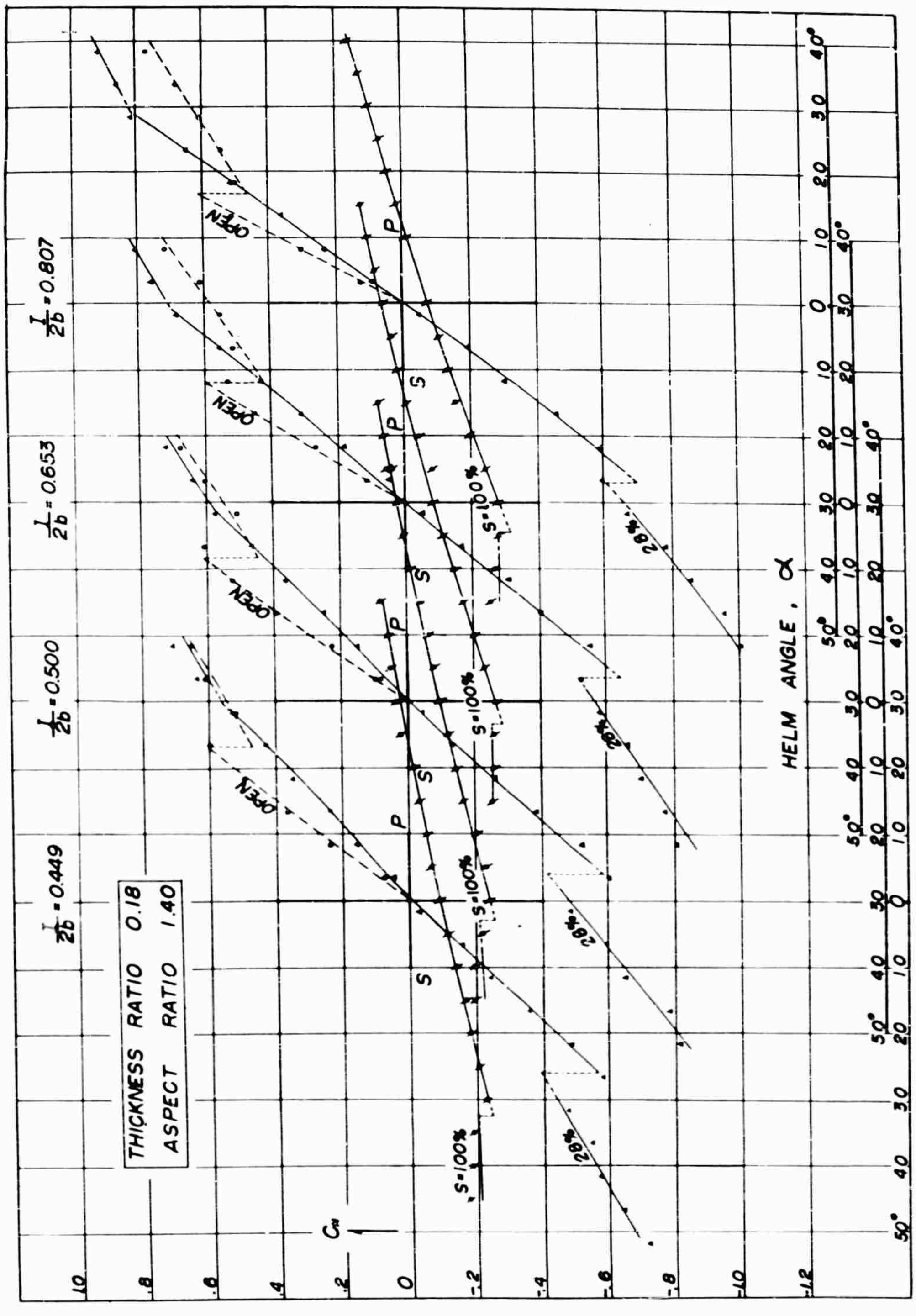

THICKNESS RATIO 0.18
ASPECT RATIO 1.40
l/2b = 0.449
l/2b = 0.500
l/2b = 0.653
l/2b = 0.807
OPEN
S=100%
28%
P
S
C_N
HELM ANGLE, α

Figure 25-2

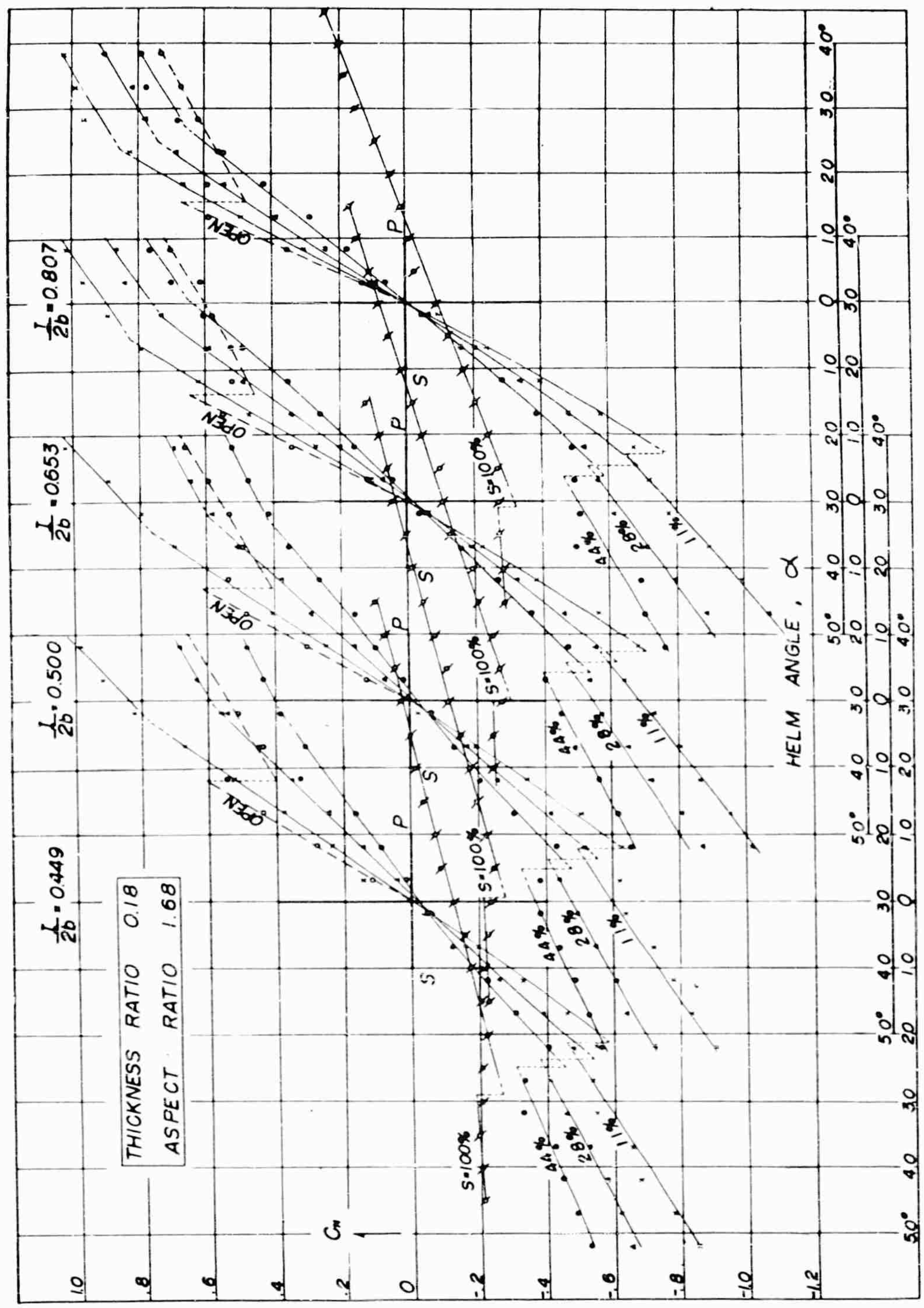

THICKNESS RATIO 0.18
ASPECT RATIO 1.68
l/2b = 0.449
l/2b = 0.500
l/2b = 0.653
l/2b = 0.807
OPEN
S = 100%
44%
28%
11%
P
S
C_m
HELM ANGLE, α

Figure 25-3

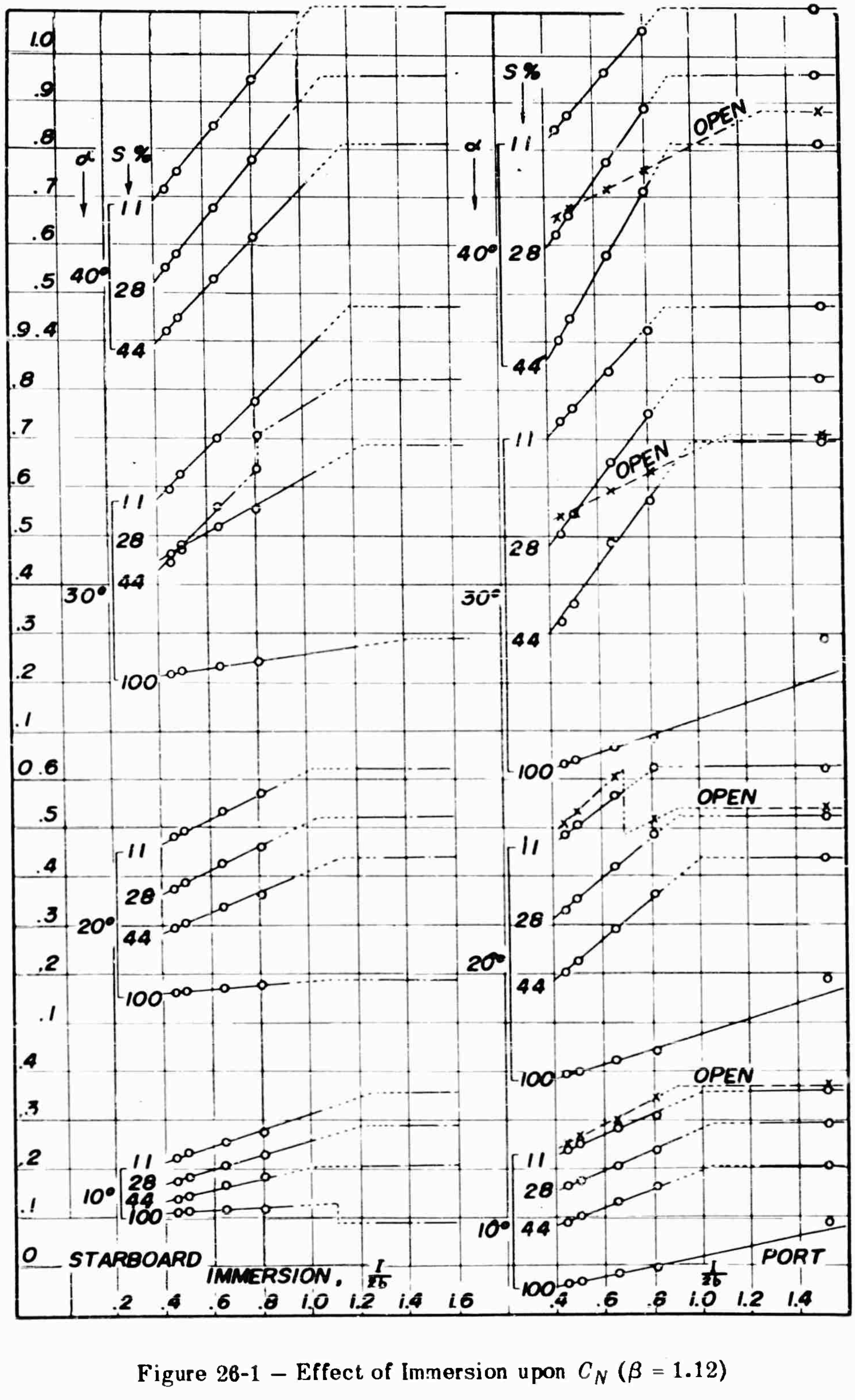


Figure 26-1 – Effect of Immersion upon C_N (β = 1.12)

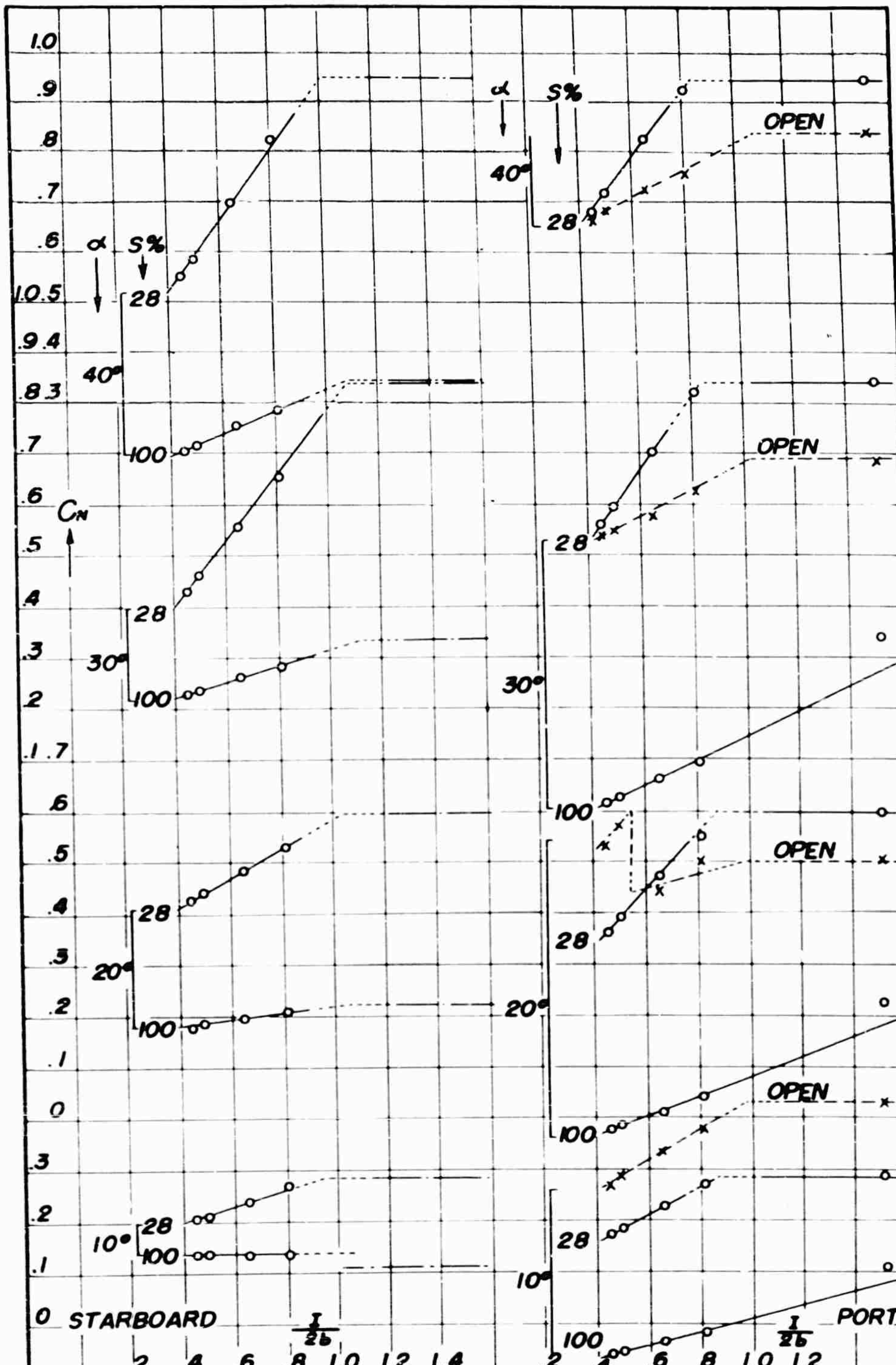


Figure 26-2 – Effect of Immersion upon C_N (β = 1.40)

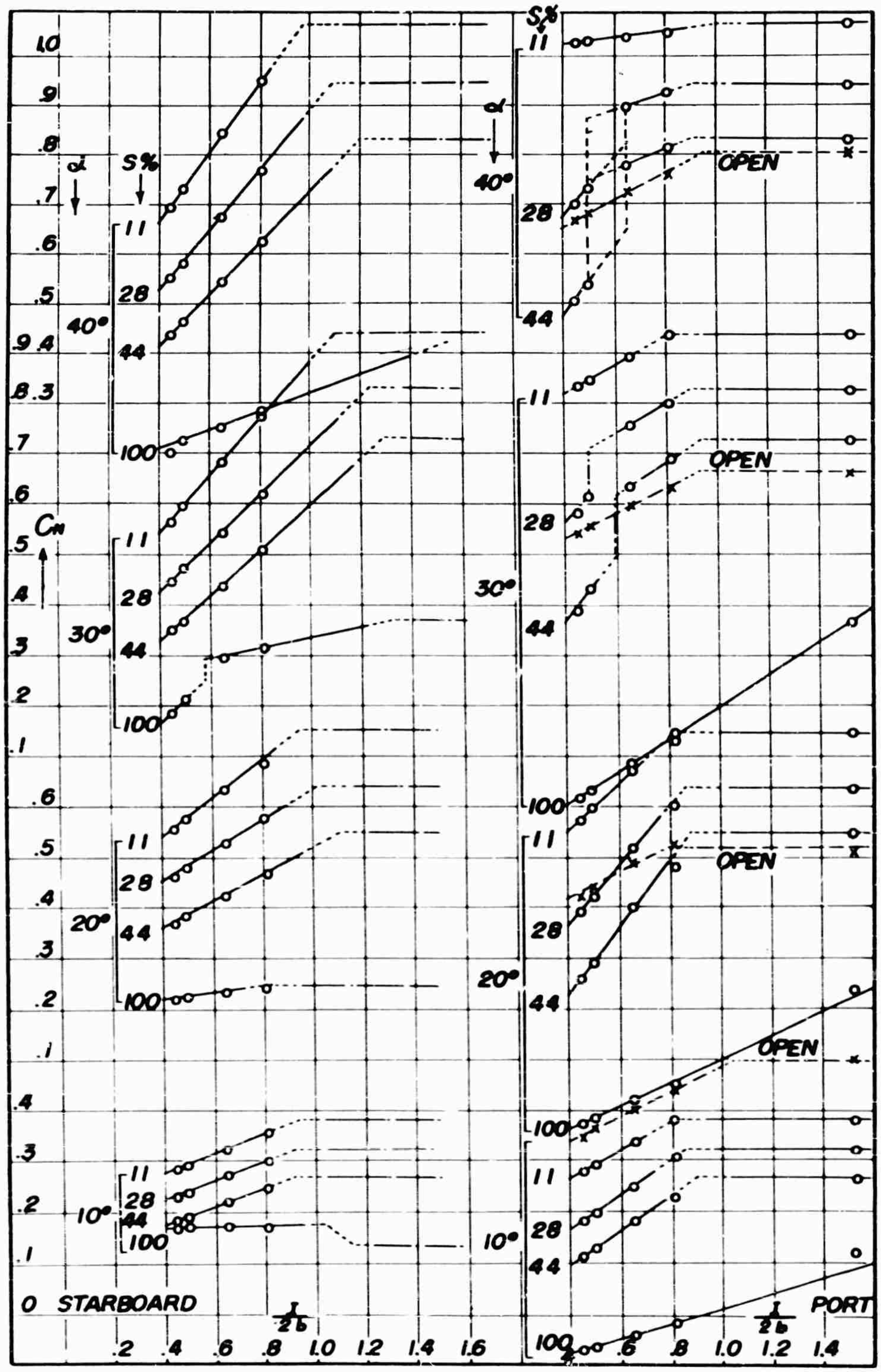


Figure 26-3 – Effect of Immersion upon C_N (β = 1.68)

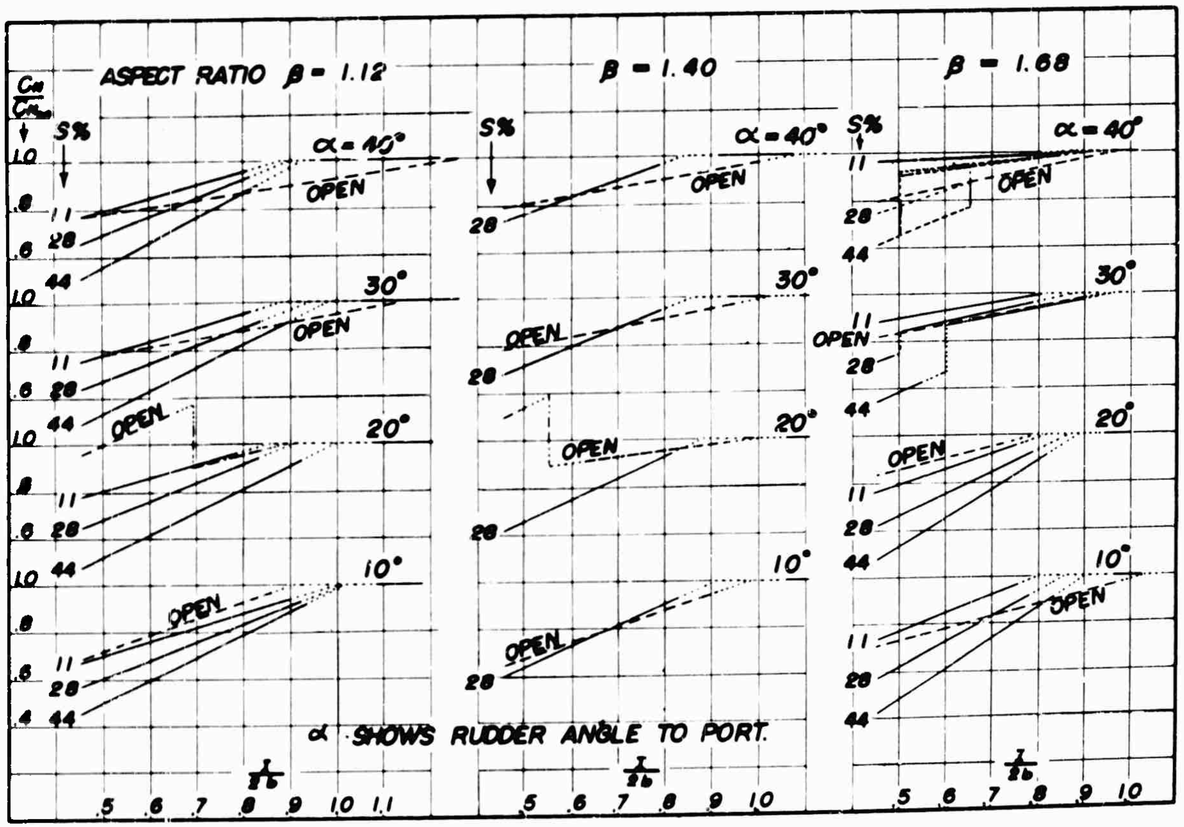


Figure 27 – Effect of Immersion upon C_N

For the general merchant ship, for example, C_N decreases by 35 percent compared with $C_{N\infty}$, if $l/2b$ is assumed to be 0.5 (trial condition) and s is assumed to be 35 percent. In short, we cannot disregard the reduction of the normal pressure of the rudder by the effect of the depth of immersion, especially when the rudder is not deeply immersed.

3.4.2 RELATION BETWEEN CENTER OF PRESSURE AND DEPTH OF IMMERSION OF RUDDER

Figure 28 shows the relation between the position of the center of pressure and the depth of immersion of the rudder, which was obtained by the experiment. The solid line in the figure corresponds to the case when the depth of immersion is large enough, and is identical with the one that is already reported in Part 2. From this figure we find that the center of pressure moves aft, according as the depth of immersion becomes smaller.

3.4.3 RUDDER TORQUE

As mentioned above, when the depth of immersion is not enough, the characteristics of the rudder deviate from that of the deeply immersed rudder. Then, how do we explain the rudder torque changes?

According to our way of expression, the normal pressure F of the rudder is shown by the following formula:

$$F = \frac{1}{2}\,\rho\,(2b)^2\;V_s^2\left(\frac{1-w}{1-s}\right)^2 \times \frac{C_N}{\beta}$$

Accordingly, the ratio of the rudder torque with that of the deeply immersed rudder is expressed as follows:

$$\frac{M}{M_\infty} = \frac{F(x - x_0)}{F_\infty\,(x_\infty - x_0)} = \left(\frac{C_N}{C_{N\infty}}\right) \times \left(\frac{x - x_0}{x_\infty - x_0}\right)$$

Here, the symbol ∞ shows the case of deeply immersed rudder.

The effect of immersion upon the rudder torque will become clear if we take an example as follows:

Taking a case of an ordinary cargo ship in trial condition and assuming that:

Depth of immersion of the rudder, $l/2b$	0.5
Aspect ratio of the rudder, β	1.6–1.8
Position of the center line of the stock from the leading edge, x_0	0.295
Slip ratio referred to effective pitch P_E, s	35 percent
Helm angle, α	30 deg

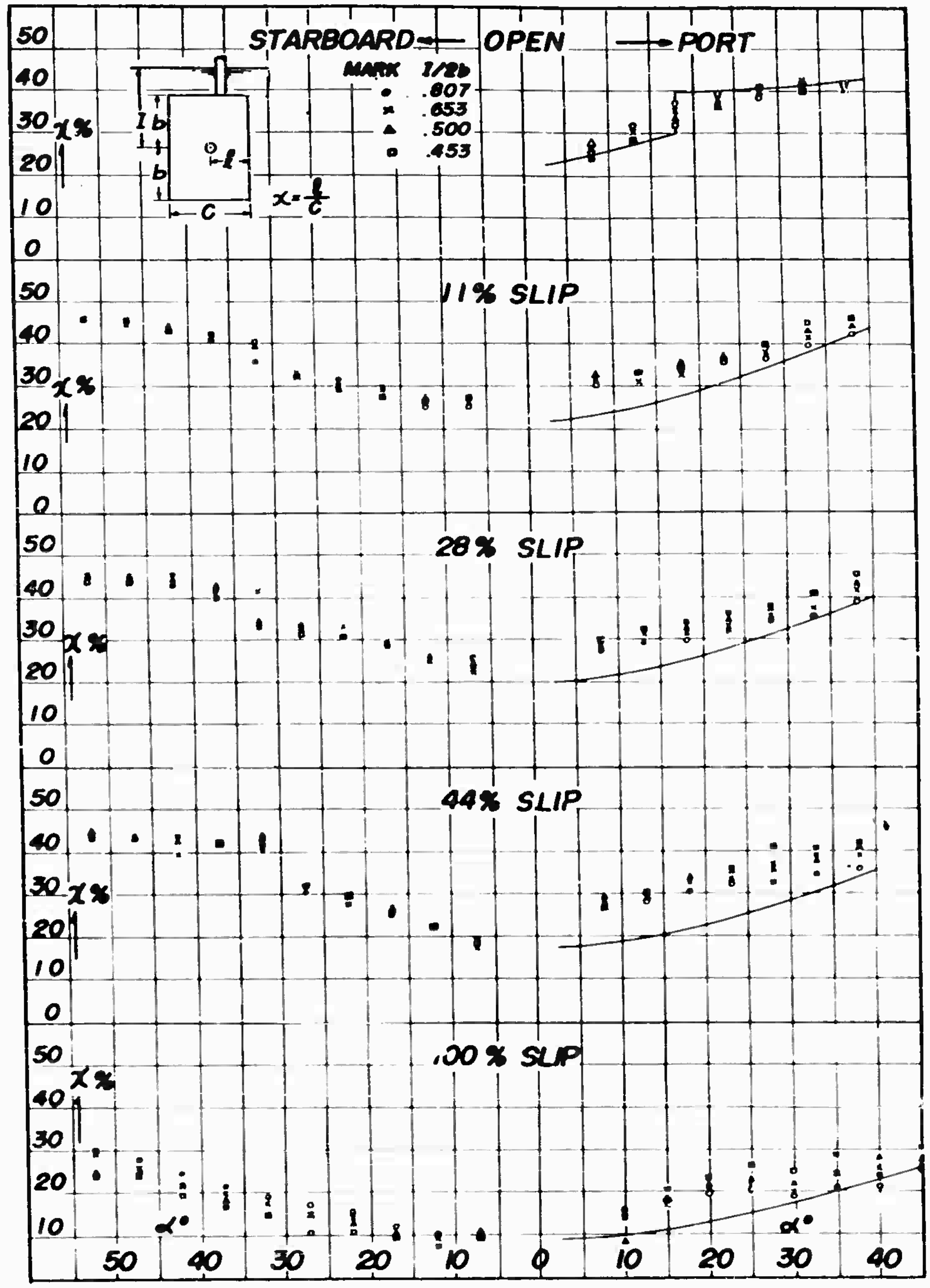


Figure 28-1 – Center of Pressure When Depth Is Varied (β = 1.12)

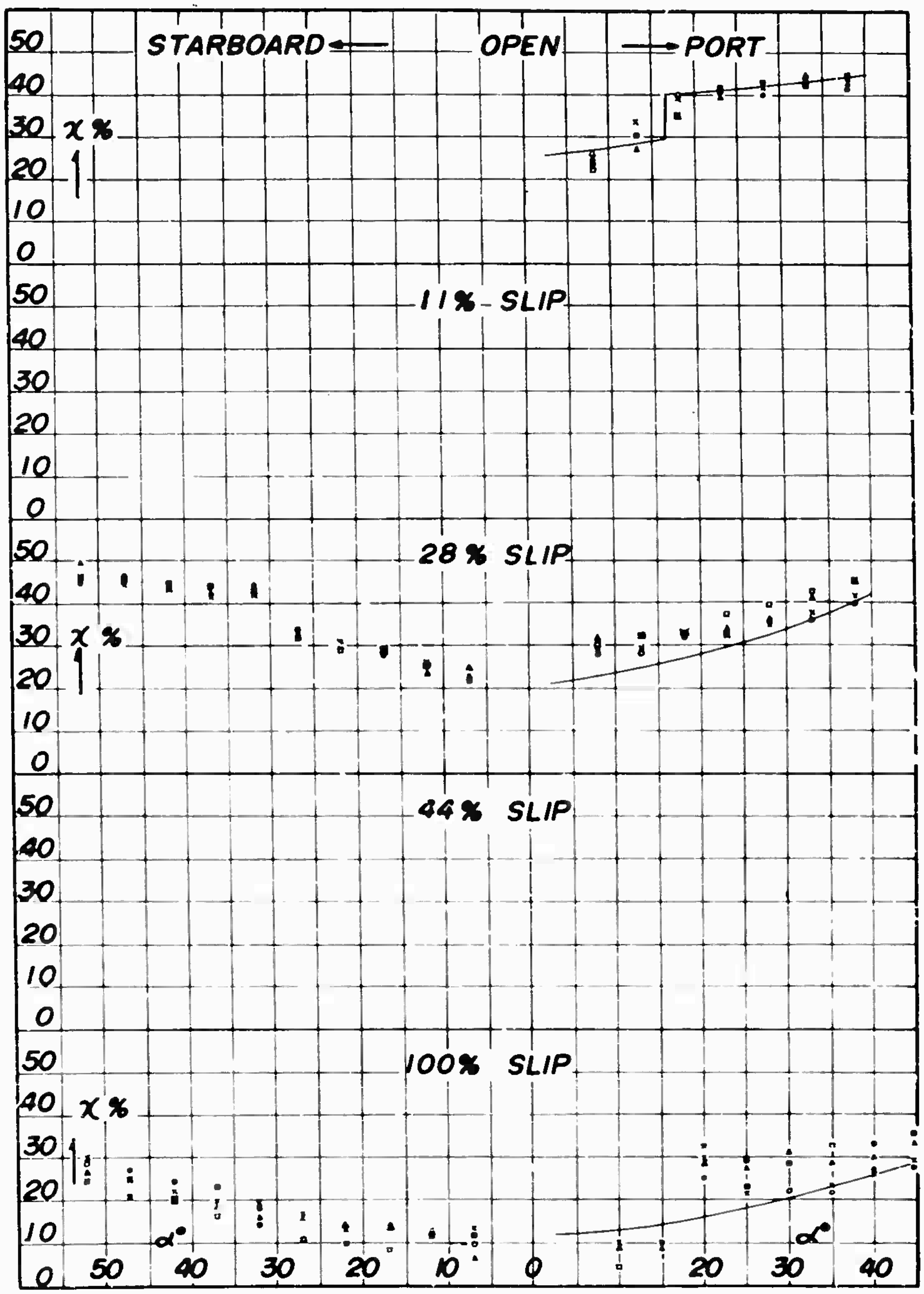


Figure 28-2 – Center of Pressure When Depth Is Varied (β = 1.40)

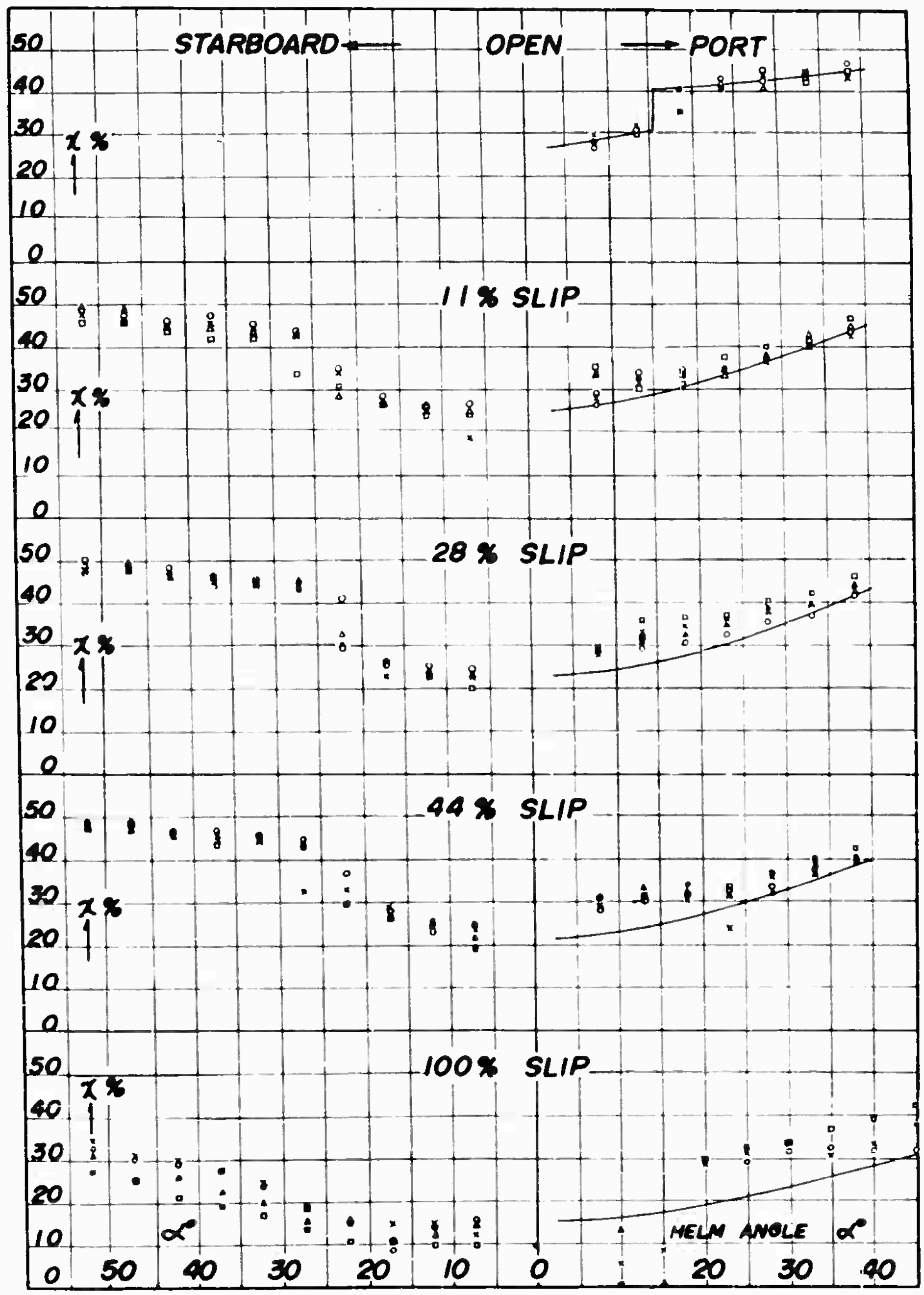


Figure 28-3 – Center of Pressure When Depth Is Varied (β = 1.68)

then we will get,

$$\begin{aligned} C_N/C_{N\infty} &\doteqdot 0.65 \quad \text{see Figure 27} \\ x &= 0.38 \quad \text{see Figure 28} \\ x_\infty &= 0.35 \quad \text{see Figure 17 (or Figure 23)} \end{aligned}$$

Therefore, the ratio of the rudder torque will become:

$$\frac{M}{M_\infty} = \frac{C_N}{C_{N\infty}} \times \frac{x - x_0}{x_\infty - x_0} = 0.65 \times \frac{0.085}{0.055} \doteqdot 1.0$$

That is, even though the characteristics of the rudder change considerably with the change of the depth of immersion, the rudder torque does not change so much, in general, when the rudder angle is large. This is because the effect of the reduction in the normal force and the increase of the distance of the center of pressure from the leading edge cancel out each other and apparently give the same torque. Now, we know the reason for the good agreement of the rudder torque of the actual ship with that of the deeply immersed rudder as we mentioned in the Introduction, 3.1.

3.4.4 EFFECT OF STEERING DIRECTION

It can be imagined that the performance of the rudder will be different when steering to starboard or port, and this will also be affected by the difference in the depth of immersion of the rudder when the rudder is working behind the screw. We can, somewhat, investigate the effect of the turning side (port or starboard) upon the performance of the rudder, using the test results shown in Figure 25, but the details about this problem will be omitted here.

GENERAL CONCLUSIONS

The most important results obtained from these three series of experiments described in Parts 1, 2, and 3, respectively, are:

1. The best rudder area exists, at least for the general merchant ship, and from the limitation of the draft, the best rudder area ratio $A/L \cdot d$ is usually about 1/45 corresponding to the aspect ratio of about 1.2.

2. Increase of the maximum helm angle is more effective for improvement of maneuverability than is increase of the rudder area only. The author believes that the maximum helm angle can be increased to about 45 deg.

3. The reason the optimum rudder area exists in the general ship is as follows: Even in case of the turning by large helm angle, the true angle of incidence of the stream to the rudder during the steady turning is considered to be very small (say, about 10 deg). The ratio C_N/β of the normal pressure coefficient to the aspect ratio, which decides the rudder force F, takes the maximum value only in case of such a small angle of incidence. The corresponding aspect ratio is about 1.2.

4. The effect of the depth of immersion of the rudder disappears when the depth of immersion $l/2b$ is about 0.9. As the depth of immersion decreases, the rudder force decreases considerably, and this tendency is larger at the larger propeller slip ratio. Moreover, the position of the center of pressure tends to move gradually aft. As for the rudder torque, these two effects cancel out each other, and consequently no apparent difference is produced by the difference in the depth of immersion.

DISCUSSIONS

S. Bindel:

I should like to present three remarks regarding Part I of this interesting paper. The first concerns the comparison between model and ship turning circles. If, for a single-screw ship or for a twin-screw and twin-rudder ship, a balance may exist between the scale effect due to the nonrespect of the Reynolds number and the scale effect due to the nonrespect of the propeller, this is not exact for a twin-screw ship with one rudder, because interactions between propeller and rudder are small. In this case we generally find that, for a given rudder angle, the turning diameter of the actual ship is smaller than the diameter of the model: the difference is often up to 10 percent.

The second remark concerns the speed reduction in turning. This reduction depends on the type of engine driving the propeller. For example, if nothing is changed on the control-gear of the engine, the torque (for a motor), or the power (for a turbine), or the rpm (for an excited-field d-c motor) may remain constant. For a valuable comparison, model experiments have to be carried out in the same conditions. This is not generally the case if the model is free; on the contrary, under a rotating arm it is possible to realize any condition.

My last remark concerns, for a single-screw ship, the difference in turning performance to port and to starboard. An explanation may be the following: generally, on a ship of this type, streamlines go up at the stern of the hull; there is an upward component of the velocity in the disk of the propeller. For a right-handed propeller, the angle of incidence on a blade and therefore the thrust is increased on starboard and decreased on port; even with a zero rudder angle, the ship then tends to turn to port.

R. N. Newton:

I thought that the authors might have given some more useful information about another parameter which affects the TD/L, and this is what we call the area of the cut-up. Now in this series of experiments this area of the cut-up does not change. Comparing one ship with another when the areas under the stern profiles are very different, there can be a large effect on this ratio TD/L. This was proven by a gentleman I mentioned yesterday, Mr. Wigley, who carried out an investigation for us many years ago. Fortunately I brought his formula along. In effect, Wigley took the results of 46 warships and analyzed them (there were 23 of them which were of similar form), and he obtained a straightforward formula from which one could obtain the TD/L very approximately, which would certainly enable the designer to compare two ships. The formula reads like this:

$$TD/L = 0.00823\,(LD-AC)\,L/A_rB.$$

I would close, in the hopes that the author will accept these comments to the paper.

H. E. Saunders:

I should like to complement what Mr. Newton has just said. I think that all of us should appreciate perhaps a little better than we do, the reasons for some of these things. Mr. Newton knows, but didn't say so, that the pressure field developed by the rudder, differential positive pressure on one side and the negative pressure on the other side, extends to the hull, so that a considerable amount of the lateral force which produces the turn is exerted on the hull, but rarely is all of it exerted on the rudder. I haven't had an opportunity of studying Dr. Shiba's paper sufficiently to know whether that was taken into account or not. Mr. Newton says, "the area of the hull in the vicinity of Dr. Shiba's rudders is constant so that whatever the factor is, it is the same except for the fact that different rudders of different sizes and laid at different angles produce different pressure fields and different pressure intensities."

Some experiments which have been run at TMB show that in a modern ship with a rudder of a little different type than this one, one-third of the total lateral pressure at the stern is exerted on the hull and only two-thirds by the rudder. I haven't made any studies, but I am convinced that in the old-time sailing ships, where the rudder was simply a narrow plank extending up and down at the stern (it couldn't be very large because one or two men had to handle it) up to 4/5 of the total lateral force was exerted on the stern of the ship and only 1/5 on the rudder. That is admittedly a guess, but almost every case we have found, regardless of the type of the hull, unless the hull is absolutely flat underneath and the rudder is separated from it at a distance, that some considerable portion of the lateral force that causes the ship to turn is exerted on the hull, not on the rudder, but by the pressure field created by the rudder.

Now to supplement what Mr. Bindel said about reasons why a single-screw ship with a right-handed propeller has to carry right rudder to handle it in a straight course and keep its heading. As pointed out by Professor Hovgaard back about 1942, the reason why the ship wants to swing with its head to port is not because of the excess thrust executed on the right-hand blade on the starboard side of the ship (and there is a great deal of excess thrust exerted on the right-hand blade) but that the differential pressure set up by those blades working at a large angle of incidence causes a greater reduction of pressure on the starboard side of the hull than it does on the port side. The reduction of pressure acts with the moments from the c.g. to the stern, whereas the increased thrust on the starboard blade acts with a very small moment. Professor Hovgaard has brought this out, and I think it is quite conclusively proved that the turning moment on the ship is developed by the greater numerical reduction of pressure on the starboard side of the afterbody than on the port side of the afterbody. The whole side of the hull is subjected to that reduced pressure and the ship wants to move over, as shown by diagrams in my book (*Hydrodynamics in Ship Design,* Vol. 1) under the section entitled Hovgaard effects. (We actually decided to name this section before Professor Hovgaard left this world.)

The fact that the ship does carry starboard rudder with right-hand propeller is important because it comes up also in the spiral test (the Dieudonné test). If you run a test on a ship with a positive stability of route, you get one graph when you plot the rate of change of heading, against rudder angle, but that graph does not pass through the zero point. In other words, it indicates that when the rate of change of heading is zero (when the ship is not swinging), two or three degrees right rudder is needed. Admiral Dieudonné may have pointed that out about 11 years ago with circulars his institution put out.

J. P. Breslin:

I wondered whether the break in the curve of normal force coefficient versus angle of attack is attributed to ventilation of the rudder, or is it due to partial separation? I would like to comment with regard to some observations which were made on a program run by the NACA (now NASA) and one which was run at the Davidson Laboratory under the NACA sponsorship on the phenomena of aeration or ventilation. (Captain Saunders has another word for this.)

My reason for bringing this up is that it has a bearing on the usability and interpretation of model data for such things. We found out that if a surface-piercing strut is run at various yaw angles, or angles of attack, it is like a rudder that is partially emerged. Struts of aspect ratios (span divided by the chord) varying from 2 down to ½ were run. (Lift coefficients versus angle of attack, α, were plotted.) During the tests at an angle of attack of about 18 to 20 degrees, we would observe almost spontaneous or rapid injection of air or ventilation. We found out that if you make a plot of the lift coefficients versus angle of attack, α, the curve increases at approximately one slope and then there is a jump down and the curve would then rise again at a different slope. In certain cases the drop would amount to about one-third of the original magnitude. The people at NACA were able to relate the occurrence of the ventilation with the appearance of separation on the section; that is, they found that if they had a dead-water region on the lee, or suction side the dead-water region behaved as though it were statically replaced by air.

The foregoing raises the question as to the legitimacy of such experiments on rudders (where one is married by the Froude numbers to operate at low speeds) because the separation phenomenon on rounded leading edges is expected to be Reynolds-number dependent, and you could get separation on a model. It is of course known that, for large aspect ratios, stalling occurs at very small angles of attack at very low Reynolds number. Luckily low-aspect-ratio rudders are not as sensitive to Reynolds number.

Moreover, another phenomenon which is not shown by Dr. Shiba's paper is that after having generated an aerated pocket if the rudder angle is then decreased the lift curve closes back (hysteresis loop) at a point (α) depending on the Froude number. As we increase the Froude number (based on the chord of the foil), the closing points move back progressively,

seemingly to approach a limit, so that perhaps, at some infinite Froude number, we might have a condition where there is no further reduction in α. So I think that with rudders which are piercing the surface, or perhaps near the surface, where there is a possibility of aeration it must be borne in mind that there are two regimes which you can operate in and the condition under which these two regimes will be obtained may well depend on the scale.

S. C. Gover:

I would like to ask Dr. Shiba if he has made any tests in which he increased the rudder area without changing the aspect ratio, in comparison with changing the area and the aspect ratio simultaneously as in these studies? In this case he kept the span constant; therefore, to increase the area he had to increase the chord. Consequently, there are two variables shown here and I am curious whether there are any material differences in the tactical diameter or normal force coefficients which might have been shown in comparative tests.

There is one short comment in regard to the unstable flow regime. In free-running model tests, we have run into this so-called rudder breakdown phenomenon. In trying to duplicate a problem, we are uncertain whether we are going to get a large tactical diameter or a small tactical diameter; in one case rudder breakdown occurs, and again under what seem to be identical conditions there is no breakdown.

A. Taplin:

I have just two questions to ask the author. One, in connection with Figure 24 which shows a comparison for various ships calculations, is very simple; are the torques on an actual ship at sea or are they on a model of a ship? The other question, the drift angle would also depend on the rate at which the rudder is turned. If the rudder is turned instantaneously, you would get virtually no drift angle. I would like to ask Dr. Shiba what rudder rate was used in these tests; the angular rate, the time it takes for the angle to go from 0 to 35 degrees?

A. Suarez:

Were any tests made to determine whether there was a region of directional instability in any one of the three vessels? Did these vessels track straight? Were they dynamically stable? I have a suspicion that these vessels were probably dynamically unstable because of their block coefficients. Is this why you had a variation of highest turning rate with increase in the block coefficient?

Author's Written Closure – Answer to Captain Saunders:

I appreciate very much your discussion. When a ship has a definite drift angle in the stationary turning, it is clear that the lateral force due to the rudder becomes very small because of the considerable decrease of incidence angle to the rudder, whereas the pressure difference due to the circulation around the hull developed by the drift angle becomes very large. Therefore it is quite natural to get different turning character with a different ship hull which has a cut-up area, for example.

Next, I would like to express a little different opinion from yours as to why a single-screw ship with a right-handed propeller has to carry right rudder to handle it in a straight course and keep heading. In such a condition, the differential pressure acting upon the hull which comes from the circulation around the hull need not be taken into consideration because of its trivial amount. As you pointed out, it is true that the reason the ship wants to swing with its head to port is not the excess thrust executed on the right-hand blade on the starboard side of the ship. This excess thrust comes from the upward flow to the propeller disk and is really very small, whereas the differential force between the upper side blade and the lower side blade is pretty large and has opposite sense compared with the above-mentioned excess thrust. Therefore this force makes the ship swing with its head to starboard and not to port.

The author believes that the reason a right-handed single-screw ship with the rudder in midship has a tendency to swing with its head to port, as you pointed out, must be attributed therefore to the differential force of the rudder (of zero angle) between the upper half and the lower half, which is very large and effective and has the opposite sense compared with the force by the propeller to swing with its head to starboard. As the horizontal incidence angle of the flow to the rudder (of zero angle) on the upper half is larger than that on the lower half due to the difference of the slip of the blade of propeller (Figure 1), the differential force between the upper half and the lower half of the rudder of zero angle is very large and makes the ship turn port side. A detailed numerical explanation is in my answer to Mr. Bindel.

Then if we test the course-keeping quality of a right-handed single-screw ship without rudder, the ship must be expected to swing with its head to starboard because of the nonexistence of this large left-turning moment of the rudder of zero angle. You may be convinced of the fact if you notice the results of the tests with the three models without rudder. As shown in Figure 2, all of them have a tendency to turn to starboard.

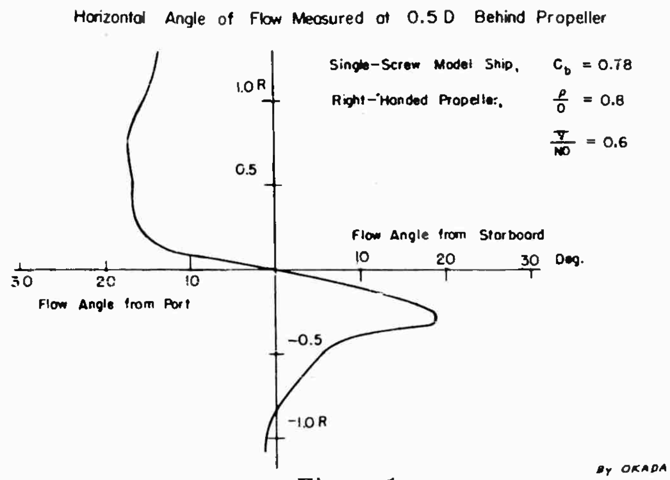


Figure 1

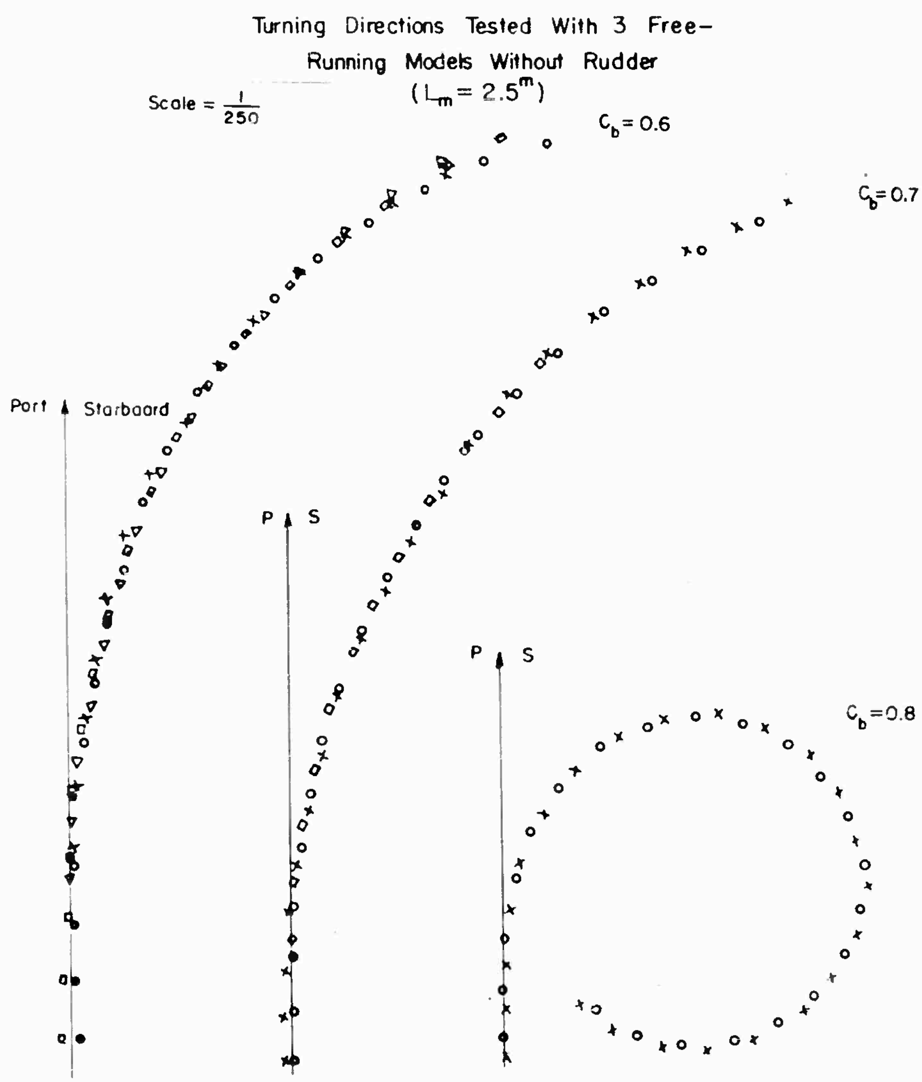


Figure 2

Answer to Mr. Newton:

The effect of the cut-up area on the turning of the ship is to be considered as one of the effects of hull forms on the turning, like the block coefficient, rake of the stem and bilge keels (dimension and position).

In this paper, however, where the optimum rudder area was its main concern, this was not discussed at all.

I appreciate very much your explanation about this problem.

Answer to Mr. Gover:

In case of the single-screw ship with single rudder, which was the main concern in this paper, if we increase the rudder area, keeping the aspect ratio constant, the lower edge of the rudder will come lower than the sole piece, and it is not practical. For this reason, I did not make this variation in the main series of the tests.

In a certain twin-screw ship with single rudder, however, I tried this variation; Figure 3 shows the results. In this case, as is clear from the relation

$$F = \frac{1}{2}\ \rho A\ V^2 C_N = \frac{1}{2}\ \rho (2b)^2\ V^2\ C_N/\beta,$$

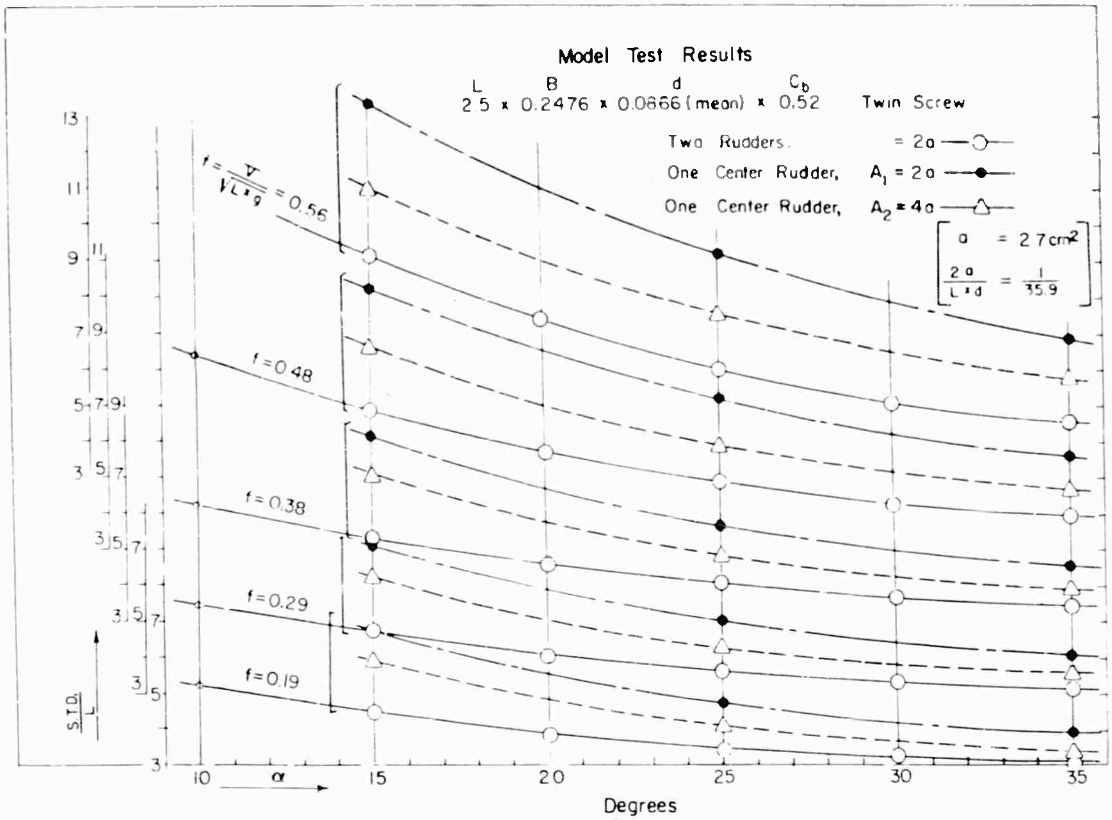


Figure 3

if we increase the rudder area, keeping the aspect ratio β constant (namely, keeping the similarity of the shape), the C_N/β stays constant; accordingly the rudder force F will increase linearly with the increase in the value $(2b)^2$, or the square of the span $2b$. Therefore the tactical diameter, advance, or the stationary turning diameter will be improved corresponding to this increase of the rudder force. However, we cannot expect so much improvement because the rudder force F or C_N is usually small while in the steady turning condition even at helm angle of 35 degrees. As a result, I would like to conclude that to increase the rudder angle would be more advantageous than to increase the rudder area.

By the experiments with aerofoils, we know very well that the stall angle is unstable and sometimes the lift coefficient curve after stalling takes a curve of quite different nature under what seems to be identical conditions.

Even at the same incidence angle, the stall angle and also the lift coefficient curve after stalling take a different angle and a different curve respectively, owing to whether we are increasing or decreasing the angle to the definite angle.

Of course this phenomenon is the same in the case of rudder. However, when the ship is going to turn by steering, the stall of its rudder is instantaneous because of the quick decrease of the incidence angle following the quick increase of the drift angle. Therefore, after the ship is transferred to the stationary turning condition, the incidence angle of flow to the rudder is much less than the unstable stall angle and the lift coefficient takes always a definite value. As a result, if we keep the condition always the same, we will get the identical value so far as the stationary turning diameter is concerned.

As for the air-drawing of the rudder, please refer to my answer to Dr. Breslin.

Answer to Dr. Breslin:

I dare say that the break in the curve in Part 2 is surely attributed to the separation and not to the ventilation (air draw). It is because the depth of the upper edge of the rudder from the free surface was always kept to be equal to the span $2b$, deep enough to avoid the air draw, in the tests. Of course, we could never observe the air draw during the tests.

I appreciate very much your discussion about the air-draw phenomenon. The author himself also started the study of this phenomenon about 20 years ago, and published a paper about the air draw of screw propeller ten years before. I would like to send you a copy of this paper and would welcome your comments on it.

In this report I made clear that for the occurrence of the air draw the separation is necessary but is not a sufficient condition, and it depends also upon the Weber's number. Also I found that the critical advance constant, that is the advance constant where the thrust drops abruptly because of the air draw, is a function of the Weber's number $W = nD\sqrt{\frac{\rho}{S}D}$, and the minimum Weber's number where the effect of the Weber's number diminishes is about 1.8×10^2 (Figure 4).

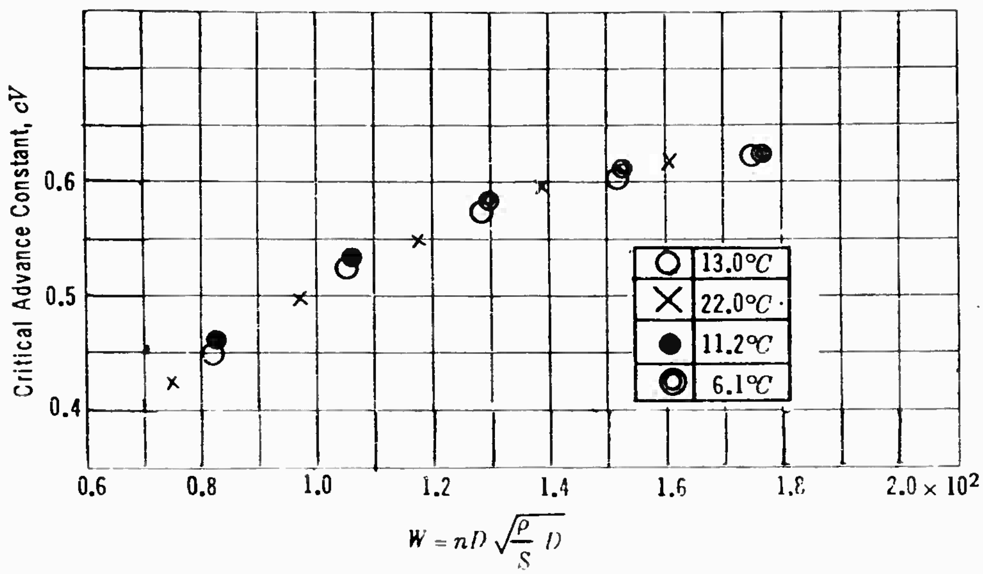


Figure 4 – Critical Advance Constant Based upon Weber's Number W'

In general, for the screw propellers of actual ships, the Weber's number is pretty much larger than 1.8×10^2; accordingly there is already no effect of the Weber's number.

For the model ships, however, usually the Weber's number for their screw propellers is less than 1.8×10^2. Therefore similarity does not exist between model and actual ship so far as the air draw of screw propellers is concerned. Namely, we need to modify or judge the results of self-propulsion model tests considering the effect of the air-draw phenomenon as well as the effect of cavitation.

The situation is almost the same in the case of the rudder tests. If we apply the results of screw propellers to the rudder, the minimum Weber's number is roughly given by

$$W = V\sqrt{\frac{\rho}{S}R} \geq 0.15 \times 10^2$$

where V is the velocity of flow to the rudder,

S is the surface tension, and

R is the radius of the leading edge of the rudder.

This means, in order to keep the similarity concerning the air-draw phenomenon, the dimension of the model rudder should be large enough to have the Weber's number larger than 0.15×10^2. Fortunately, this condition is satisfied if we adopt the model ship larger than about 5 meters long, which is practically possible. As the air draw of the rudder is closely related to the separation, we have to be careful, especially in the case where the rudder is steered to a large helm angle by which the separation is apt to take place. However, even when the helm

angle is large, the drift angle of the ship increases rapidly right after the rudder is settled. As a result, the incidence angle of the flow to the rudder decreases also remarkably. Accordingly, even if the air draw were induced by the separation, it is instantaneous or lasts for only a very short period of time (Figure 22-1, Figure 24 in my paper) and will vanish pretty soon following the disappearance of the separation. Therefore in the stationary turning, we need not worry about the scale effect by the difference of the Weber's number between model and ship because of the nonexistence of the air draw.

The scale effect due to the difference of the Reynolds number exists on the rudder as well as on the hull. However, as is seen from the NACA report on the aerofoil, that effect is very small while the incidence angle of the flow to the rudder is very small. Accordingly, we can assume that the effect of the Reynolds number is negligibly small for the rudder while the ship is turning stationarily in which the incidence angle of the flow to the rudder is small.

These are shown in the fact that the results of model tests agree very well with the results of the actual ship, even though we use the small model of 2.5-meter length, as far as the stationary turning diameter D/L is concerned

Considering the facts that the effect of the Weber's number and that of the Reynolds number can be neglected in the stationary turning, we adopted 2.5-meter model ships for convenience.

Answer to Mr. Taplin:

The curves in Figure 24 show the rudder torque and steering velocity measured on the sea for several actual ships. Of course, the turning angles of the ships were measured simultaneously. The drift angle, right after the helm angle was settled to 35 degrees, was generally about 3~6 degrees. The circle marks show values of rudder torque just after the steering was over, estimated through the data acquired in this series of model tests. In doing this, as the drift angle is modified by the propeller, an angle of 33 degrees $[= 35° - (3° - 1°)]$ was used.

As for the model ships, the time it takes for the angle to go from 0 to 35 degrees was adjusted to take always about 1.6 seconds.

The effect of the time is now under experiment. However, we need not worry about it, so far as the stationary turning is concerned. As you know, my main object of study is to find the best rudder area and to make clear why the best rudder area would exist.

Answer to Mr. Suarez:

In order to investigate the course stability or straight-course-keeping quality, I made a plot as shown in Figure 5, by taking the value L/D versus rudder angle from Figure 5 in my paper. The value L/D at rudder angle zero in this curve can be considered as an index of

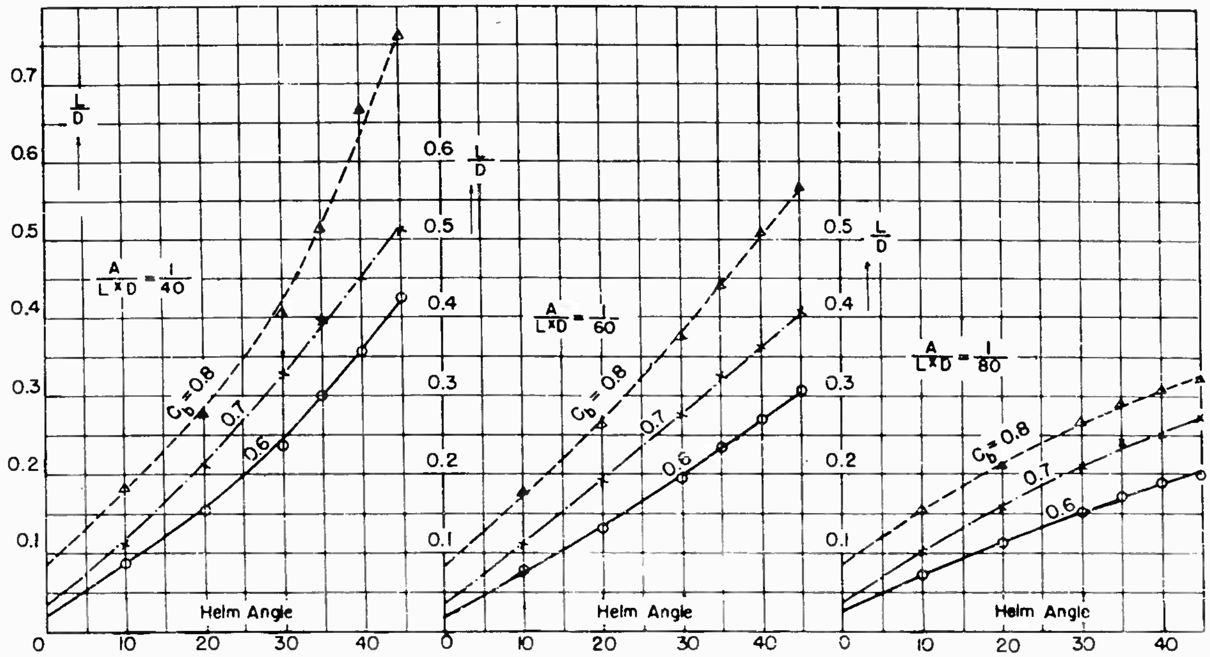


Figure 5

course unstableness. All of the curves show that values of L/D at zero rudder have more or less definite positive values. In other words, all of these hulls are not stable in course and cannot go straight.

From the amount of this L/D value, we can judge the degree of course stability for twin-screw ships.

For a single-screw ship, however, we cannot use this value of L/D at zero rudder as the criterion of course stability. The reason is that, for a single-screw ship, as there are upward flow and vertically varied wake distribution, the propeller gives two turning moments of opposite sense and also the rudder gives another turning moment—for example, with a right-handed propeller there are left turning moments by the propeller due to the upward flow, right turning moments by the propeller due to the vertically varied wake distribution, and also the left turning moment by the rudder of zero angle due to the difference of the horizontal incidence angle by the position in the upper and lower part of the rudder. Therefore it is not clear whether the value of L/D at zero rudder is due to the unstability of the ship hull or to the above-mentioned turning moments due to the propeller and the rudder.

However, as you find in my answer to Mr. Bindel's discussion, the left turning moment by the right-handed propeller due to the upward flow is negligibly small compared with the right turning moment by the propeller due to the variation of the wake. (See Figure 2. These

are the results tested with three models without rudder, all of which indicate a right-hand turning tendency.) The left turning moment due to the rudder of zero angle is remarkably larger compared with either of them. As a result, the right-turning screw ship has the tendency always to turn to port side. From this fact, for single-screw ships, we cannot simply consider the L/D value of zero rudder as the index of course unstableness.

According to the increase of the block coefficient, the value of L/D at rudder of zero angle increases gradually. This is natural, because the resistance to the turning decreases gradually with increase of the fullness.

To discuss this problem of course-keeping quality further, we need some tests planned for this purpose. The author has made a special device to give the pure turning impulsive moment to the free-running model and tried to investigate this character precisely; however, he failed to get the necessary turning angle because of the short capacity of that device, and the author has to say that the course-keeping quality for these models is not yet clear. These three types of models are, however, the representative hull form of ships constructed in Japan in present days, and we do not hear about any trouble of their course-keeping quality. As for the reference, I will show you the standard of the rudder area ratio for Japanese single-screw ships.

1/64 for ordinary merchant ships. 1/72 for supersized oil tanker.

This shows that actually there are no objections to using a smaller rudder area with the increase of the block coefficient because of the decrease of the turning resistance and because of the trivial increase of difficulties in course-keeping qualities.

Answer to Mr. Bindel:

1. Usually because of the difference of the wake fraction by the scale effect, the number of revolutions of the propeller of the actual ship which sails straightforward is 1~3 percent higher than that of the self-propelled model modified by the friction correction. During the turning also, even if we could succeed in getting the proper amount of friction correction in model tests (that is still unknown in present days), the number of revolutions of the propeller of the actual ship is expected to be a little higher than that of the model.

In the single-screw ship with a single rudder, as the velocity of the flow to the rudder is about $(1 - w)\ V + sNP = NP$, the rudder force in the actual ship is larger than that of the model ship by the difference of the number of revolutions. This means a smaller turning diameter for actual ships than for the model.

However, if we do not execute the friction correction for the self-propelled model in the turning test (actually the amount of this correction is not yet clear in want of data), the number of revolutions of the model propeller becomes a little higher, and as a result, the rudder force increases; therefore the turning diameter cannot be considered always larger than that of the actual ship. The author tried to check this, and found that these effects almost cancel each

other, fortunately, and the turning diameter was affected very little at all by the scale, in general. This report is mainly concerned with the case of single-screw ships.

In case of a twin-screw ship with single rudder, the rudder is not affected so much by the ship stream of the propeller as is a single-screw ship, where the rudder force is governed by the velocity of $(1 - w)\ V$. Accordingly, its turning diameter is always smaller in the actual ship than in the model regardless of whether friction correction is added or not. The author has no definite idea whether the amount of this difference is 10 percent or not, as he has never tried the comparison tests about this point. However, generally in twin-screw ships, the block coefficient is rather small and accordingly the wake fraction is small. Accordingly, we can consider that the difference in turning performances between ship and model may be not so large.

2. As you pointed out, the speed reduction while the ship is turning depends pretty much upon the character of the main engines. The results in this paper were those of models driven by the d-c shunt motor, as is reported in the paper. Accordingly, if we assume that the advance constant does not vary by the small change in the number of revolutions, the turning speed will be reduced according to the decrease of the number of revolutions. In order to investigate this difference of the character of the main engine, the author also tried to use the small internal combustion engine; however, it was not stable in its performance so he could not get the satisfactory results.

3. Lastly, I would like to answer about the straight-course-keeping quality of ship. For example, in case of the single-screw ship with a right-handed propeller, even at zero rudder, there are three factors to induce the moment to swing the ship's head as follows:

The first is the effect of the upward oblique flow at the position of the propeller.

The second is the effect of the variation of the wake distribution along the vertical direction at the screw disk.

The third is the effect of the unbalance of the lateral force of the rudder (of zero angle).

The first will give the ship a moment to turn to port, the second to starboard, and the third will give a moment to turn to port. (About the wake effect, please see Figure 1 in my answer to Captain Saunders.)

The author roughly estimated the amount of those moments as follows:

When the propeller works in nonuniform or oblique flow, the relative advance speed to the flow and the angular velocity of the blade element varies with the position while it is turning. For this reason if we take the average during one revolution, a certain amount of force and moment to a certain direction will be produced by the propeller.

Suppose that there is the flow of velocity u to the negative direction of y-axis, the angular velocity Ω of revolution of the propeller varies partially as follows:

$$\delta\Omega = -\frac{u \sin \chi}{r}$$

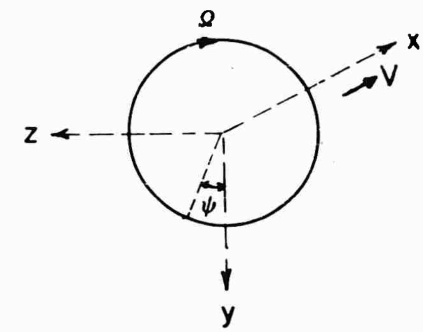


Figure 6

From this variation, the propeller will produce a moment M around the y-axis.

$$M = -\frac{uT}{\Omega}\left(1 - \frac{J}{2K_t} \times \frac{dK_t}{dJ}\right) \qquad [1]$$

When we express the nonuniform flow on account of the difference in wake fraction by the position by

$$\delta V = h\frac{\gamma}{R}\cos\chi$$

then a force Z will act on the propeller on the direction of z-axis, and the moment will be

$$lZ = -\frac{hQ}{VR}\left(\frac{J}{2K_q} \times \frac{dK_q}{dJ}\right) l \qquad [2]$$

(Nippon Zosen-Kyokai No. 83.)

Here V = advance speed of propeller
R = radius of propeller
Q = torque of propeller
T = thrust of propeller
$K_t = \dfrac{T}{\rho N^2 D^4}$, $K_q = \dfrac{Q}{\rho n^2 D^5}$
n = R.P.S.
$J = \dfrac{V}{nD}$
$D = 2R$
l = distance between propeller and c.g. of ship

The valu, expressed by Equation [1] is the moment to turn the ship to port on account of the upward oblique flow to the propeller, and Equation [2] shows the moment to turn the ship to starboard on account of the difference of the wake fraction by the position.

For the 2.5-meter length model, if we assume

v = 1.0 m/s (ship speed)

w = 0.25 (mean wake fraction)

J = 0.5

$\rho = \dfrac{2.5}{2}$ m

u = 0.27 v (corresponding to 15 deg oblique flow)

$$\frac{dK_t}{dJ} = -0.36, \quad \frac{dK_q}{dJ} = -0.038, \quad K_t = 0.18, \quad K_q = 0.026$$

we can estimate roughly

M = − 0.0017 kgm

lZ = 0.018 kgm

Namely, the moment to turn to starboard on account of the nonuniform flow of the wake is much larger than the moment to turn to port on account of the upward oblique flow. Now the horizontal incidence angle of flow to the rudder behind a screw is known in Figure 1 (Answer to Captain Saunders). Assuming this relation as shown in Figure 7 for convenience, the author estimated the value of turning moment of the rudder (of zero angle) applying the blade element theory as

$$Fl = -0.102 \text{ kgm}$$

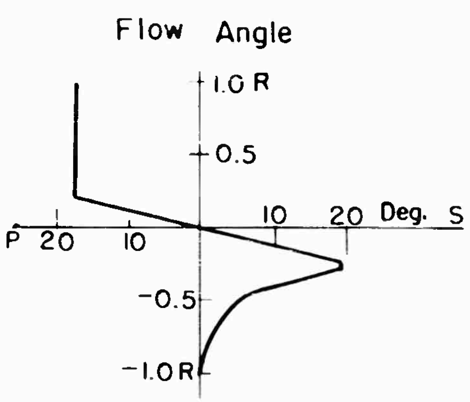


Figure 7

Negative sign means port turning.

Namely, the author concludes that the moment from the rudder (of zero angle) is the most predominant among these three, and the result is that the single-screw ship swings its head to port. As to the turning direction when the single-screw ship is without its rudder, please refer to my answer to Captain Saunders.

When the rudder is steered to a certain angle and there is the upward oblique flow, we have to think that there is already a circulation around the ship's hull. Accordingly the turning moment of the hull must be taken into consideration; however it does not play so big role in the turning of the ship when the ship is keeping its head almost straight by a very small helm angle.

NOTES ON RUDDER DESIGN PRACTICE

by

A. Taplin

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Department of the Navy

ABSTRACT

A design method for rudder calculations is presented. This method, now in use in the Bureau of Ships, is an aerodynamic approach to determination of lift, drag, bending moment, and twisting moment. A numerical example is worked out for a spade rudder, including allowances for aspect ratio, sweep angle, taper ratio, and effective speed and angle of attack. This design example includes a method for going from rudder hydrodynamic torque to steering gear torque. In addition, the paper discusses other practical considerations in rudder design, such as the materials commonly used for all-welded and for socketed connections of rudderstock to rudder, rudder plating, and framing; design features of antifriction and sleeve bearings for rudderstock; and design of bearing seals for rudderstocks.

INTRODUCTION

A combination of rudder and hull can be designed to provide good turning, course-keeping ability, and ability to initiate and check rapid swinging. Specific numerical criteria for these qualities are available in Reference 1.* Also a comprehensive analysis is available in Chapters 37 and 74 of Reference 2 and in Reference 3 for determining rudder location, area, and planform.

This paper describes the rudder design work that follows after the rudder location and shape have been preliminarily selected. Such design work involves computing the hydrodynamic forces and then determining structural and mechanical features for reliability with low cost.

Although these are not official standards, they represent current practices by designers in the Bureau of Ships.

RUDDER FORCES, TORQUES, AND MOMENTS

In the computation of rudder forces, bending moments, and torques, we use aerodynamic methods plus allowances from experience. For this discussion, cavitation is excluded. In the simplest terms, we convert a ship rudder into a more-or-less equivalent free-stream control surface for which we have wind-tunnel data. This procedure will be explained for the case of a spade rudder in a propeller race.

FLOW SPEED AND ANGLE OF ATTACK

One of the early steps involves finding the speed and angle of attack for the ship rudder. Figure 1 shows model test data for the velocity field at the rudderstock of a twin-screw, twin-rudder destroyer, taken from Reference 4. This represents the ship going ahead in a straight

*References are listed on page 139.

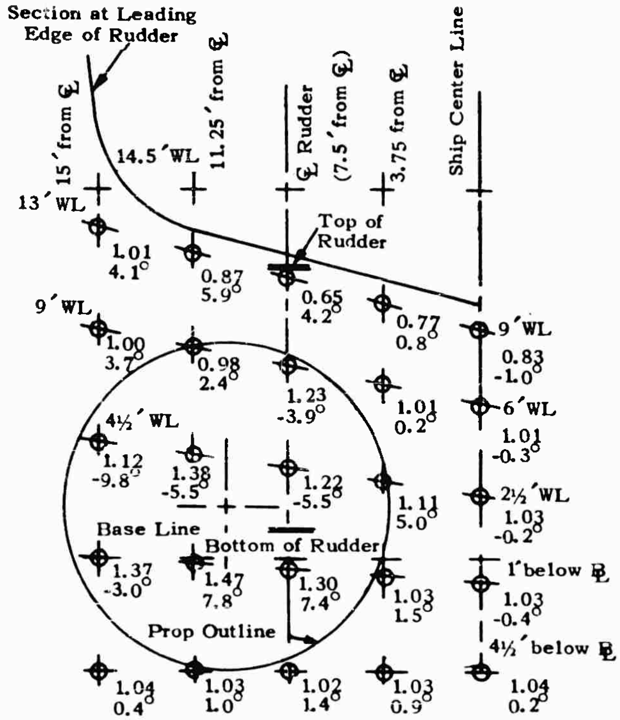

Figure 1 – Velocity Field at Rudderstock
for a Twin-Screw, Twin-Rudder Ship

line at constant speed, with positive angles indicating inward flow. Even for this special case, it is clear that the upper part of the rudder is in boundary-layer flow, and the lower part in propeller outflow. Both the magnitude and the direction of velocity are varying along the rudder span. We know from full-scale trials of USS NORFOLK (DL 1),[5] that maximum forces occur about 10 sec after the rudder has started moving. By then the ship has begun to swing. Thus a drift angle, speed reduction, and transverse flow set in, and the complex flow of Figure 1 becomes even more involved. To handle this we make two simplifying assumptions:

1. For a rudder behind a propeller, the assumption that speed of water over the rudder is 80 to 100 percent of (Ship Speed) (1 + Propeller Slip). This is rather conservative, and results in rudder forces which are not likely to be exceeded. The choice between 80 percent and 100 percent is a matter of judgment, based on how much of the rudder is in the propeller race.

2. The assumption that the effective attack angle is 5/7 of the actual rudder angle. This allows for drift angle at the rudder, and is based on trial experience and the convenience of 5/7 as a coefficient. Thus, in considering a 35-deg rudder angle we would enter the wind-tunnel data at 25-deg attack angle. (See Lines 1 and 2 of tabulations in Appendixes A and B.)

EFFECTIVE ASPECT RATIO

One more "equivalent" is needed before we can use free-stream data, and that is the effective rudder aspect ratio. In ideal cases it is span squared over area, and it is doubled if there is little or no gap between the rudder and the hull. If we consider that the hull gap is small at zero rudder, and large at full rudder, we compute effective aspect ratio by multiplying geometric aspect ratio by a coefficient varying linearly from 1.9 or 2.0 at 0-deg rudder to 1.0 at full rudder angle. If the gap is considered large at all angles, we use the effective aspect ratio equal to the geometric throughout.

USE OF WIND-TUNNEL DATA

A detailed example is shown in Appendix A. The tabulation form permits checking step-by-step. The source of wind-tunnel results is TMB Report 933, Reference 6. This report covers aspect ratios 1.0, 2.0, and 3.0 for the NACA 00XX sections now used. The specific steps involved are:

1. Use the charts for quarter-chord sweep angle $\Omega = 0$ deg. For the desired angle of attack and effective aspect ratio, cross-fair and interpolate to get corresponding values of lift and drag coefficients C_L, C_D, and chordwise center of pressure $(CP)_C$. Enter the values in the tabulation.

2. Repeat for sweep angle of 11 deg.

3. Interpolate between these 0-deg and 11-deg figures to get values for the actual quarter-chord sweep.

4. If the taper ratio (defined as tip chord/root chord) differs materially from the 0.45 values of TMB Report 933, a lengthy taper ratio correction must then be made. This is shown in detail in Appendix B.

The significant conclusions are that making this particular taper ratio correction changes the 35-deg rudder angle coefficients as follows:

1. The normal force coefficient increases from 0.810 to 0.928, per line 20.

2. What is more important is that the fore-and-aft center of pressure measured from the mean chord leading edge has moved from 0.266 (mean chord) to 0.319 (mean chord), per line 27. This changes torque lever arms very greatly. If the rudderstock were at the quarter-chord point, the original lever arm would be 1.6 percent of the mean chord, compared with a corrected lever arm of 6.9 percent of the mean chord.

The calculations shown in Appendixes A and B result in the basic hydrodynamic loads on the rudder. Rudderstock location can be adjusted to get a desired balance of positive and negative torque.

RUDDERSTOCK SIZE

The computation of rudderstock size is straightforward, and is based on the resultant force F_R, spanwise center of pressure, and bearing locations. The bending moment is a maximum at the point of zero shear. For conventional engineering accuracy, this can be taken at the middle of the lower bearing. (If refinement is desired, the stress distribution can be computed, and a more precise zero shear location obtained.) The spanwise center of pressure is available in TMB Report 933, and can usually be selected accurately enough without interpolation. The combined stresses from bending and torsion are then computed by standard methods, such as

$$\text{Combined Stress} = \frac{\text{Flexural Stress}}{2} + \sqrt{\left(\frac{\text{Flexural Stress}}{2}\right)^2 + (\text{Torsional Stress})^2}$$

With loads calculated by this aerodynamic method, we permit stresses up to 50 percent of yield. (Formerly, with the less sophisticated Joessel formula, we limited stress to 40 percent of yield.)

STEERING GEAR TORQUE

Going from hydrodynamic torque Q_H (at the rudderstock just outside the hull), to steering gear torque (at the key between rudderstock and tiller), involves two adjustments:

1. An error allowance Q_A, for chordwise center of pressure. This is taken as the torque from normal force times about ± 2 percent of the mean chord. If our ship rudder is quite similar to the wind-tunnel model, we would use an error allowance of ± 1 percent; if the contrary, we might allow more than ± 2 percent. This allowance is quite important for balanced rudders with the stock near the quarter-chord point, and is practically negligible for unbalanced rudders. The allowance is a measure of the effect on torque of all our uncertainties, such as variations in Reynolds number, roughness, thickness, effective aspect ratio, etc. The values of $\pm Q_A$ are added to Q_H, as shown in the torque curves of Appendix A. This converts the single line Q_H to a band.

2. A bearing friction allowance Q_F. To derive this, we first compute rudderstock bearing reactions. These are the reactions to the external resultant rudder force at the spanwise center of pressure, combined vectorially with reactions to any internal tiller force. The frictional torque at each bearing is then the resultant reaction multiplied by the coefficient of friction and the bearing turning radius. The upper and lower bearing friction are combined as $\pm Q_F$, and added algebraically to Q_H and $\pm Q_A$, as shown in Appendix A. This allows for cases where the steering gear is driving the rudder and where the rudder is tending to overhaul the steering gear.

PRECISION OF FORCE AND TORQUE CALCULATIONS

In general, we feel that the method described here predicts rudder forces conservatively and with reasonable accuracy. We still have a long way to go, however, before we can predict rudder torques accurately. Some of the difficulty is probably inherent in ship construction, since:

1. Sister ships sometimes show considerable variation in ram pressure for identical rudder movement.

2. Where torque is measured at the rams by hydraulic pressure and in the rudderstock by strain gages, as in Reference 5, there are unexplained discrepancies.

We maintain a continuing comparison of the actual versus predicted steering gear torques. We expect that we will be able to reduce our margins or allowances for torque by using this basic aerodynamic approach coupled with some additional research and analysis of trial data.

BASIC CONSTRUCTION FEATURES

In constructing and supporting a rudder, modern practice permits a choice in two structural features:

1. The method of connecting the rudder and rudderstock.
2. The type of bearings.

Although these may be treated independently, they are related to one another and to hydrodynamic features. These are discussed below for spade rudders constructed of steel.

CONNECTION BETWEEN RUDDER AND RUDDERSTOCK

The rudder can be welded integrally to the rudderstock as shown in Figure 2, or secured mechanically as shown in Figure 3. A comparison indicates the advantages and disadvantages of each type as follows:

1. Rudderstock Material

Rudderstocks suitable for welding are practically limited to a yield strength of about 30,000 psi. These stocks are unalloyed, nonheat-treated, with carbon about 0.25-percent maximum and manganese about 0.75-percent maximum. Where welding is not required, we can use either higher carbon steels or heat-treated alloy steels (typical analysis 0.28-percent carbon, 0.32-percent manganese, 0.54-percent chromium, 3.4-percent nickel, 0.06-percent vanadium, and 0.40-percent molybdenum). With these we get yield strengths of from 45,000 to 100,000 psi.

Figure 4 compares the diameters of equal strength solid stocks for a range of yield strength. For example, the diameter for 30,000-psi yield is 1.44 times that for 90,000-psi yield. The reduced diameter of the high yield stock results in reduced frictional torque at the bearings, hence in reduced steering gear size. The reduced weight of the high yield stock tends to offset its greater cost per pound, as well as reduce ship weight.

2. Rudder Thickness

The all-welded rudder is generally thicker, since it "swallows" a larger diameter stock. The maximum rudder thickness to avoid separation and cavitation is not known exactly. We try to limit maximum thickness to about 26 percent of the chord.

3. Shipping and Unshipping

The all-welded one piece combination requires more clearance below and also greater crane capacity for shipping and unshipping.

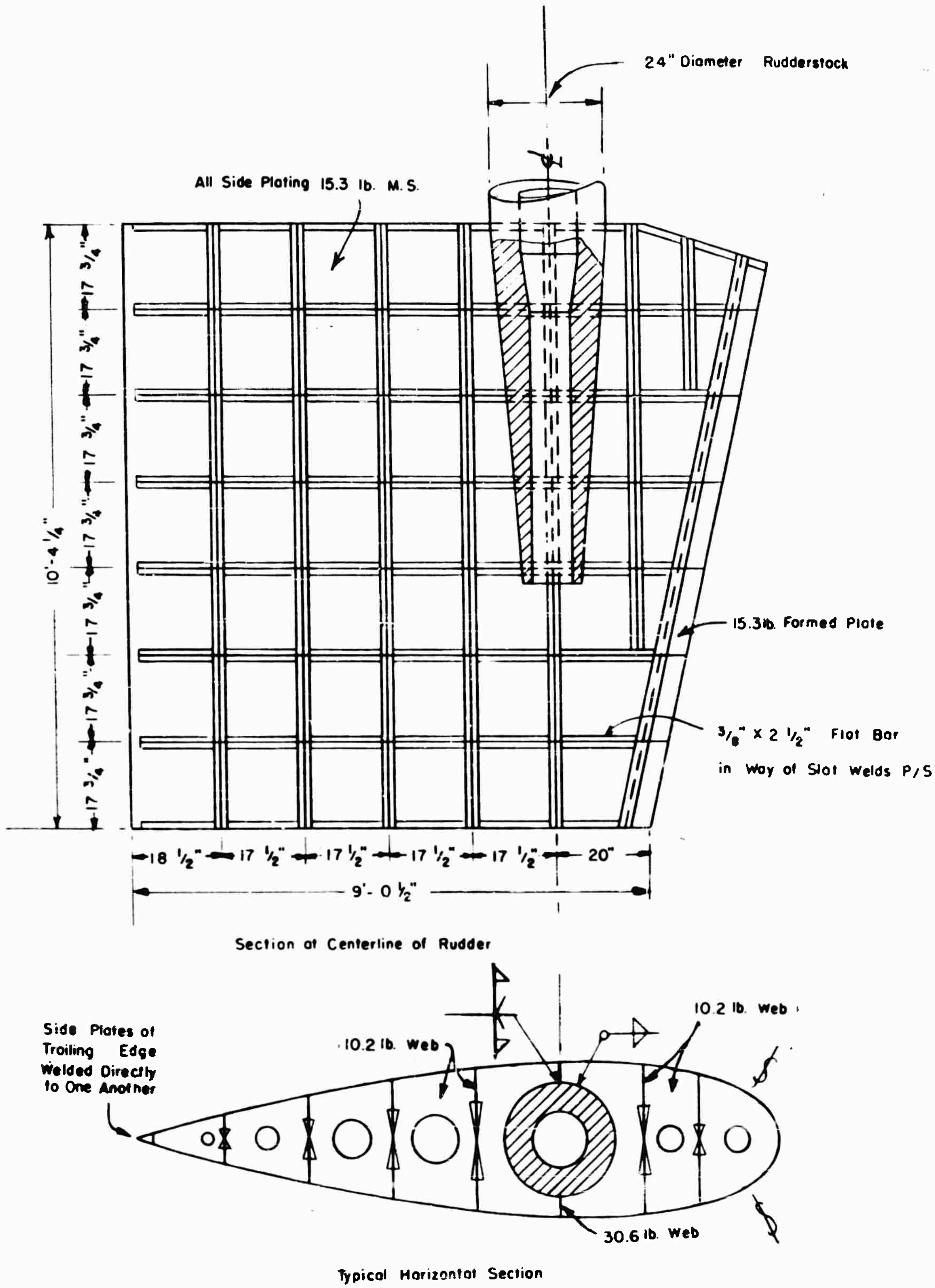

Figure 2 – Rudder and Stock Welded Integrally, Medium-Speed Ship

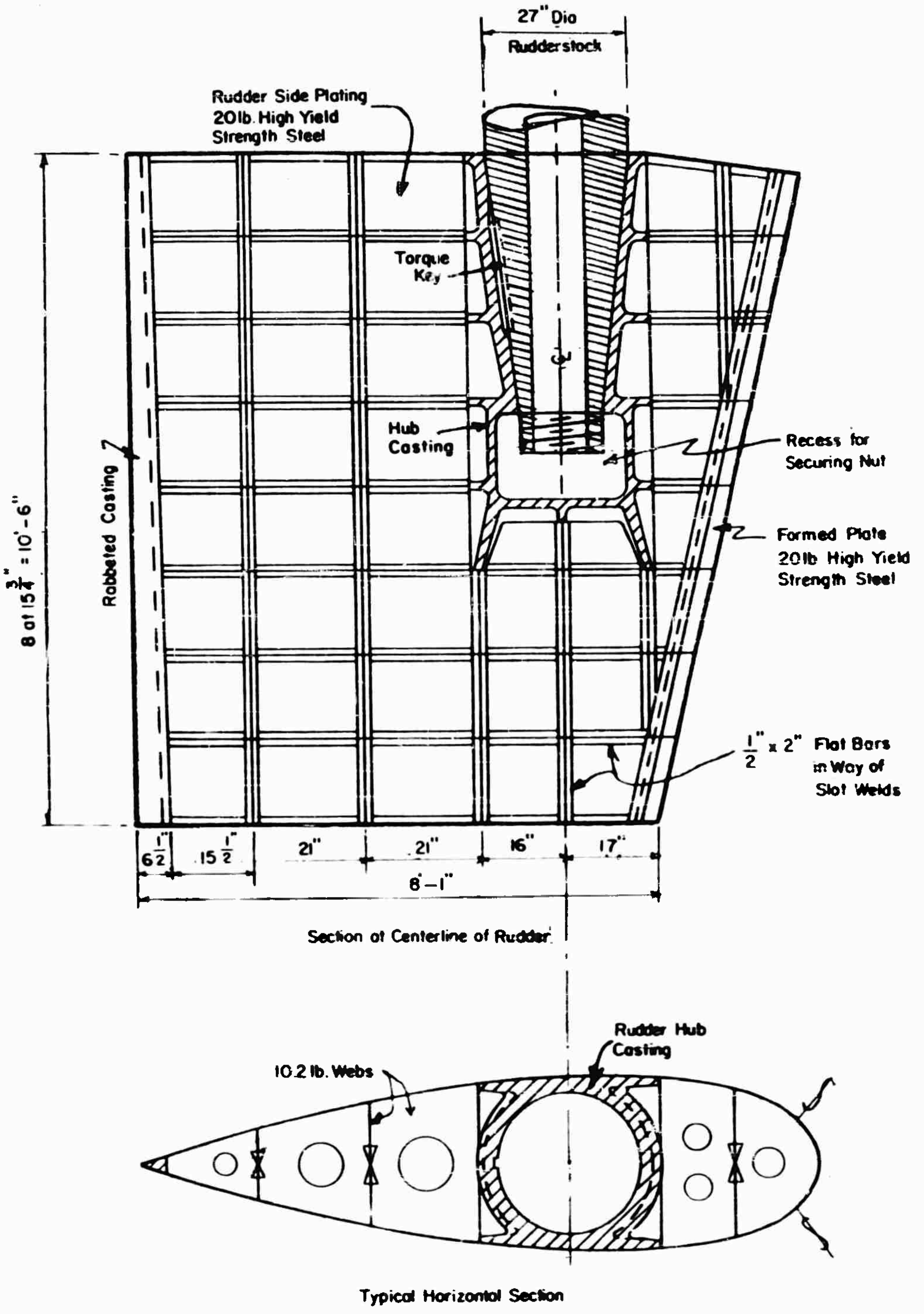

Figure 3 – Rudder and Stock with Securing Nut, Rudder in High-Speed Propeller Race

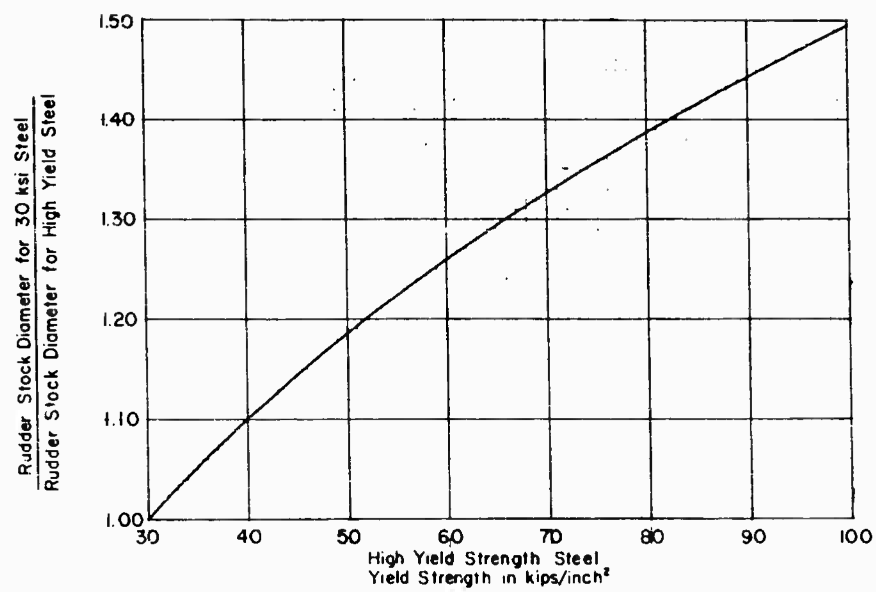

Figure 4 – Rudderstock Diameters for Steels of Various Strengths

4. Fabrication Cost

The all-welded combination is cheaper and easier to fabricate. It does not require the precise machining and fitting of the tapered connection between rudderstock and rudder hub.

The final choice as to the connection between rudder and rudderstock is made with the foregoing features in mind. Sometimes the rudder planform is altered to get an acceptable solution to these construction problems.

RUDDER BEARINGS AND SEALS

SLEEVE BEARINGS

Sleeve-type bearings for rudderstocks are commonly made of reinforced laminated phenolics (known by trade names such as Marine Micarta, Tufnol, etc.) or metals suitable for sea water (manganese bronze, gun metal, Stoody metal, etc.). These bearings are not harmed by sea water, so that special or adjustable seals at the shell are not required. A simple sand excluder is usually provided at the hull. We generally estimate the coefficient of friction at 0.20 for this slow-turning (around ½ rpm) intermittent operation.

ANTIFRICTION BEARINGS

Antifriction bearings for rudderstocks are of the ball, taper-roller, or spherical type, such as shown in Figure 5. There is a problem in providing a hull seal, and two types are used: (1) a Syntron-type seal, adjustable only in drydock, and (2) an adjustable gland which requires setting the hull bearing a foot or two above the shell. The great advantage of

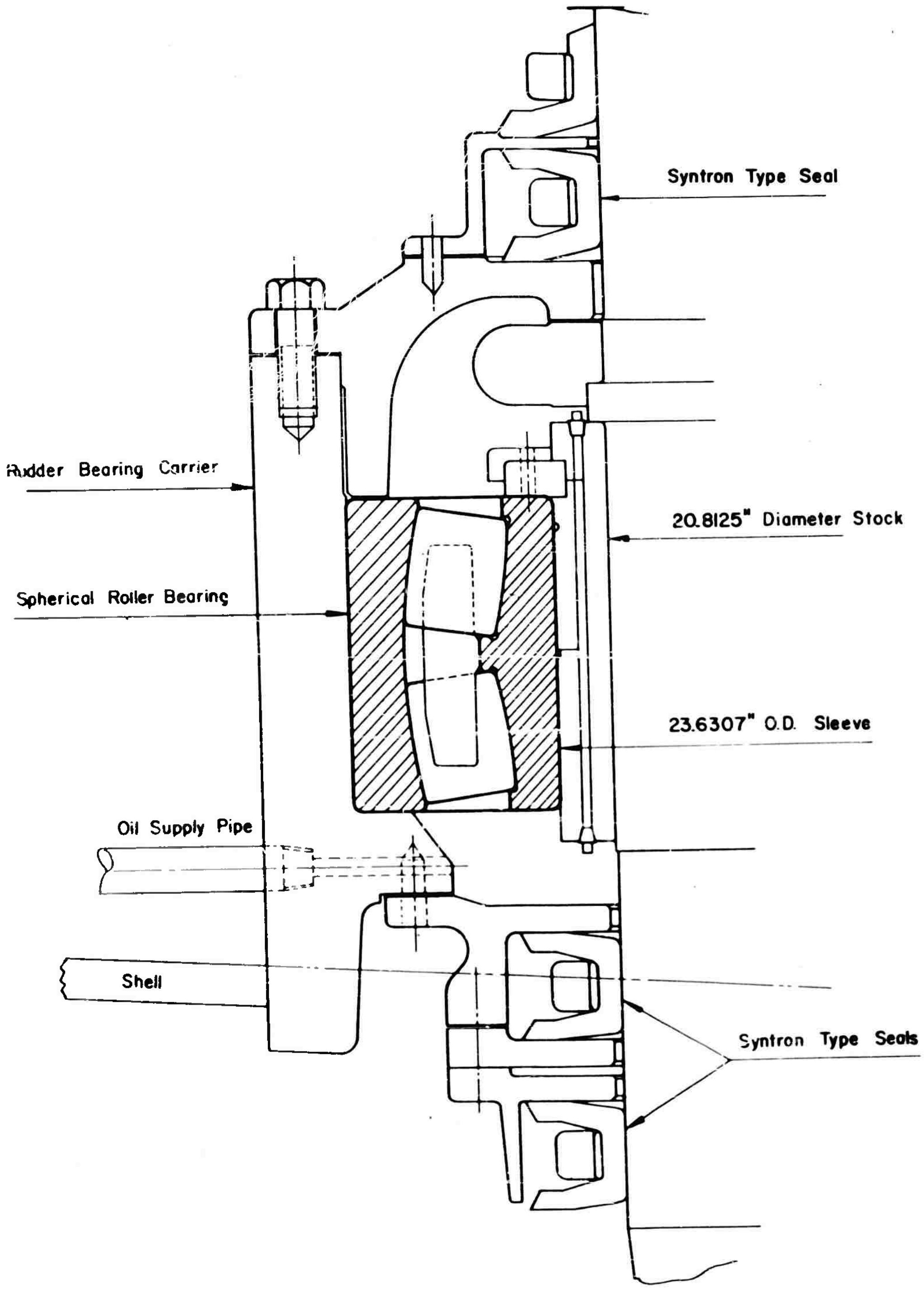

Figure 5 – Antifriction Bearing for Rudder Stock

antifriction bearings is the low coefficient of friction. We use 0.010 or 0.015. This considerably reduces steering gear size as compared with a sleeve bearing installation. (The previous description of bearing friction calculations permits numerical evaluation of this.) There had been some concern over possible Brinneling or work-hardening of rollers. The thought was that on long high-speed voyages with little rudder movement, only one or two rollers would get many cycles of loading from the propeller. This has not happened, however.

The choice between sleeve and antifriction bearings is difficult. Initial cost, including the effect on the steering gear size, is one factor. Maintenance, involving the ability to get replacement parts in an emergency, is another factor, and favors the use of sleeve bearings. The Navy uses both types of bearings, depending upon the particular ship application.

RUDDER PLATING AND FRAMING

Rudder plating and framing practice for ships is still quite empirical, unlike airplane practice for control surfaces. We have tried aerodynamic techniques, with the rudder as a stressed-skin structure subjected to lifting surface loads. The scantlings, however, would be far lighter than for structures we know have not stood up well in service. Presumably, therefore, some other intense loadings are involved, such as propeller tip cavitation bubbles collapsing on the rudder, or the propeller tail cone vortex impinging on the rudder. Fatigue and stress corrosion of panels of plating might also be involved, since propeller blade rate results in millions of cycles of loading.

Our plating and framing practices as shown in Figures 2 and 3 are accordingly based on the results of service experience. We select plating thickness and locate internal webs in rudders so that panels are limited in span to about 50 thicknesses of plating. Rudders in high-speed propeller races are usually plated with a tough, strong steel (an 80,000- or a 100,000-psi yield steel with about 2-percent nickel and 1-percent chromium), rather than the mild steel used otherwise. This practice, based on a study of rudder casualties made by Captain Saunders, has been working out well.

One desirable incidental result of using the NACA 4-digit symmetrical series (OOXX) is that the trailing edge has a natural half-breadth, rather than coming to a feather edge. This provides more ruggedness which is useful for astern operation.

VIBRATION

We have been fortunate in having had only one rudder-induced vibration problem in the last few years. This was on the USS FORREST SHERMAN (DD 931) Class, and the previously mentioned flow survey, Figure 1, was part of the investigation. For this twin-screw, twin-rudder ship, 4-cps hull vibration became objectionable at high speeds. The rudders were set with trailing edges 3 deg inward, for optimum propulsion. During sea trials, the Boston Naval Shipyard's vibration team noticed that vibration was considerably reduced when the ship

ordered a few degrees of rudder angle. A new systematic set of sea trials was then conducted, using turnbuckle steering gear links to get a range of initial rudder settings. It was found that setting the rudders with trailing edges 1½ deg outward eliminated the objectionable vibration, and this change was made in all ships of the class. References 4 and 7 give a detailed account of the investigations made by the David Taylor Model Basin towards explaining what was happening. The studies are not yet complete, but it appears that there was synchronism between rudder torsional frequency and transverse hull vibration.

To reduce chances of rudder-induced vibration, we generally do the following:

1. Compute transverse frequency of the rudder plus rudderstock. If there is synchronism with shaft rpm or propeller blade rate, we make changes in the configuration.

2. Clearance between the rudder and propeller is kept in line with past successful practice.

3. Model flow tests are conducted in TMB's circulating-water channel, and erratic flow is corrected. This sometimes involves changing the hull lines aft.

ACKNOWLEDGMENTS

The design practices described here have been developed over several years by many people. However, I particularly wish to acknowledge the contributions of Mr. R.K. McCandliss and Mr. S. Cauldwell of the Electric Boat Division, of Dr. L.F. Whicker and Mr. A. Goodman of the David Taylor Model Basin, and of Mr. H. Lo of the Bureau of Ships.

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APPENDIX A

COMPUTATION OF RUDDER FORCES AND TORQUES

GIVEN: A spade rudder as shown in Figure 6 is in the race of a propeller. At maximum ship speed of 30 knots, the propeller slip is 17.2 percent. The rudder is close enough to the hull so that full reflection (double the geometric aspect ratio) can be assumed at 0-deg rudder angle; also assume linear decrease to 1.0 times geometric aspect ratio at full rudder angle of 35 deg. The rudder has NACA OOXX sections and square tips. Roller bearings are used on the rudderstock.

TO FIND: Forces and torques throughout the range of rudder angles.

PROCEDURE:

Rudder area = 11.35 ft × 9.01 ft = 102.3 sq ft

$$\text{Taper ratio} = \frac{\text{tip chord}}{\text{root chord}} = \frac{5.59 \text{ ft}}{12.43 \text{ ft}} = 0.45, \text{ so}$$

that NACA 0015 curves of TMB 933 apply without taper ratio correction.

The step-by-step procedure is tabulated below. In this example, subscript 1 refers to data taken directly from TMB 933, and subscript 2 refers to desired data. Additional information, where the tabulation is not self-explanatory, is:

Line 2. Take effective angle of attack = 5/7 rudder angle.

Line 3. The geometric aspect ratio is $\frac{(\text{span})^2}{\text{area}} = \frac{(11.35)^2}{102.3} = 1.26$

Use 1.26 at 25-deg attack angle, and prorate other angles for 2 × 1.26 at 0-deg attack angle.

Line 4. Reynolds number for ship, based on rudder mean chord =

$$\frac{(56.4 \text{ ft/sec})(9.01 \text{ ft})}{0.000015 \text{ sq ft/sec}} = 34 \times 10^6$$

This is about ten times greater than the highest Reynolds number test in TMB 933. Use the highest Reynolds test values in TMB 933 as being the closest. Obtain lift coefficient C_L by interpolating and fairing from Figures 45, 60, and 67 of TMB 933 for sweep angle Ω = 11 deg.

Line 5. Similar to Line 4, except for Ω = 0 deg use Figures 44, 55, and 66.

Line 6. Straight line interpolation for the desired sweep angle Ω = 9½ deg.

Line 7. Similar to lines 4 and 5, except read drag coefficient C_D, and no interpolation is needed.

Line 8–11. Lift and drag are used in the conventional aeronautical sense of forces normal to and in line with the flow. Lines 10 and 11 are the normal components of lift and drag coefficients.

Line 12. The normal force coefficient, for use in computing hydrodynamic torque, is Line 10 plus Line 11.

Line 13. Interpolate from Figures 45, 60, and 67 of TMB 933 to get the chordwise center of pressure, aft of mean chord leading edge.

Line 14. Interpolate from Figures 44, 55, and 66.

Line 15. Straight line interpolation between Lines 13 and 14 for the desired sweep angle of 9½ deg.

Line 18. The sign convention is that used for aeronautical control surfaces. Plus values indicate moments tending to drive the rudder to larger angles. Minus values indicate moments tending to restore the rudder to 0 deg.

Line 19. Propeller race speed = (1 + slip) (ship speed) = (1 + 0.172) (30 × 1.69 ft/sec) = 59.4 ft/sec

Estimated effective speed of flow over rudder = 95 percent × 59.4 = 56.4 ft/sec

$$q = \text{Unit dynamic pressure} = \frac{\rho}{2} v^2 = \frac{1.99}{2} \frac{\text{lb sec}^2}{\text{ft}^4} (56.4 \text{ ft/sec})^2$$

$$= 3170 \text{ lb/ft}^2$$

$$Sq = \text{Dynamic pressure on rudder} = (3170 \text{ lb/ft}^2)(102.3 \text{ ft}^2)$$

$$= 325{,}000 \text{ lb}$$

To get Line 19, multiply 325 kips by the normal force coefficient from Line 12.

Line 20. Line 19 times Line 18.

Line 21. This is the arbitrary error allowance of 1½ percent of mean chord. The mean chord is 108.12 in., from Figure 6.

Line 22. This is the torque error allowance, Line 21 times Line 19.

Line 23. The resultant force coefficient is $\sqrt{(C_L)_2^2 + (C_D)_2^2}$

Line 24. The resultant force is Line 23 times 325 kips. (See also explanation for Line 19.)

Line 25. The spanwise center of pressure at 25 degrees attack angle is, from TMB 933, about 49 percent span, or (0.49) (11.35 ft) = 5.562 ft from root chord. From the given bearings locations, the spanwise CP is then 5.562 ft + 1.021 ft = 6.583 ft below the centerline of lower bearing. Assume a double ram steering gear, which applies torque without side force. Using the resultant force F_R from

Line 24, the upper bearing radial load $F_U = F_R \dfrac{6.583 \text{ ft}}{6.333 \text{ ft}} = 1.039\ F_R$

The lower bearing radial load $F_L = F_R + 1.039\ F_R = 2.039\ F_R$

By separate calculation, the radii to center of rollers are: $R_U = 8.65$ in. and $R_L = 14.5$ in. Use coefficient of friction = 0.01. The total frictional torque is then $0.01\ [F_U R_U + F_L R_L] = 0.01\ [(1.039\ F_R)\ (8.65 \text{ in.}) + (2.039\ F_R)\ (14.5 \text{ in.})] = 0.386\ F_R$

Line 25 is then Line 24 times 0.386.

Lines 26–29. These involve addition of the error and friction allowances to get an envelope of torque as shown in the plot of "Final Torque Curves," Figure 7.

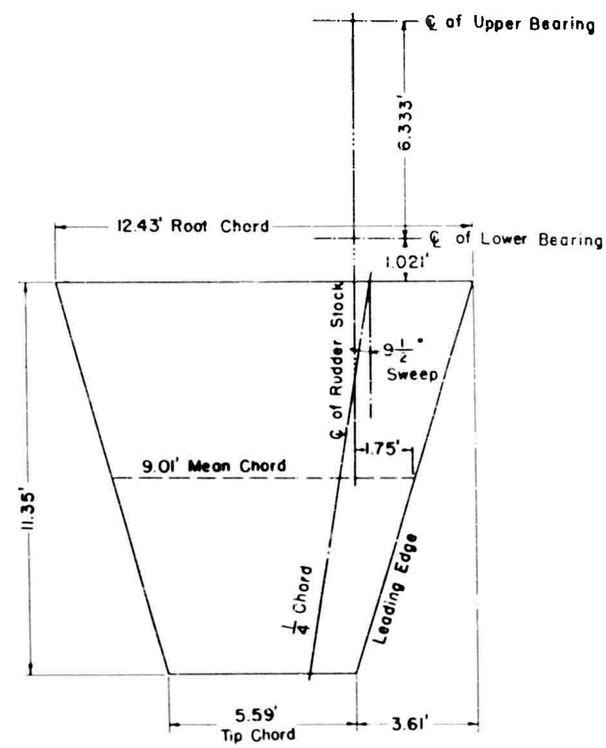

Figure 6 – Rudder Outline and Bearings Locations

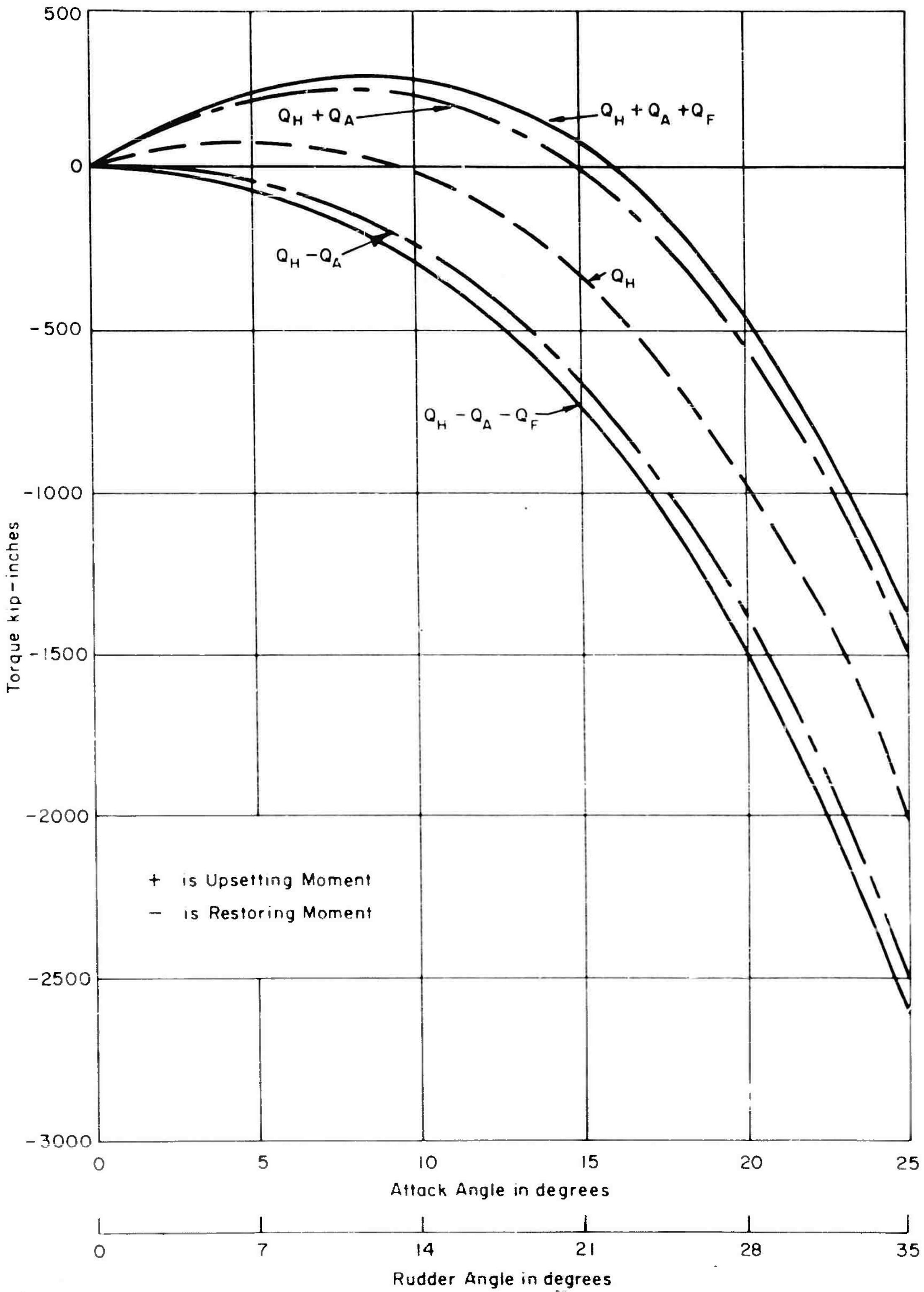

Figure 7 – Final Torque Curves

TABULATION OF CALCULATIONS FOR APPENDIX A

Line						
1	Rudder Angle, deg	7	14	21	28	35
2	Angle of Attack α, deg	5	10	15	20	25
3	Effective Aspect Ratio	2.27	2.01	1.76	1.51	1.26
4	C_{L_1} at $\Omega = 11°$	0.236	0.448	0.630	0.792	0.919
5	C_{L_1} at $\Omega = 0°$	0.228	0.430	0.615	0.775	0.908
6	C_{L_2} at $\Omega = 9\frac{1}{2}°$	0.235	0.446	0.628	0.790	0.918
7	$C_{D_1} \cong C_{D_2}$	0.015	0.043	0.088	0.154	0.244
8	$\cos \alpha$	0.99619	0.98481	0.96593	0.93969	0.90631
9	$\sin \alpha$	0.08716	0.17365	0.25882	0.34202	0.42262
10	$C_{L_2} \cos \alpha$	0.2341	0.4392	0.6066	0.7424	0.8320
11	$C_{D_2} \sin \alpha$	0.0013	0.0075	0.0228	0.0530	0.1031
12	C_{N_2}	0.2354	0.4467	0.6294	0.7954	0.9351
13	$(CP)_{\bar{C}}$ from LE at $\Omega = 11°$	0.1835	0.1945	0.2090	0.2295	0.2556
14	$(CP)_{\bar{C}}$ from LE at $\Omega = 0°$	0.1865	0.1944	0.2072	0.2246	0.2492
15	$(CP)_{\bar{C}}$ from LE at $\Omega = 9\frac{1}{2}°$	0.1839	0.1945	0.2088	0.2288	0.2547
16	$(CP)_{\bar{C}}$ from LE at $\Omega = 9\frac{1}{2}°$, in.	19.88	21.03	22.58	24.74	27.54
17	CL Stock from LE, in.	21.00	21.00	21.00	21.00	21.00
18	Torque Arm, in.	+ 1.12	− .03	− 1.58	− 3.74	− 6.54
19	Normal Force F_N, kips	76.5	145.2	204.6	258.5	303.9
20	Q_H, kip-in.	+ 86	− 4	− 323	− 967	− 1988
21	Allowance Torque Arm, in.	1.62	1.62	1.62	1.62	1.62
22	Q_A, kip-in.	124	235	332	419	492
23	Resultant Force Coefficient	0.235	0.448	0.634	0.805	0.950
24	Resultant Force, kips	76	146	206	262	309
25	Q_F, kip-in.	29	56	80	101	119
26	$Q_H + Q_A$, kip-in.	+ 210	+ 231	+ 9	− 548	− 1496
27	$Q_H - Q_A$, kip-in.	− 38	− 239	− 655	− 1386	− 2480
28	$Q_H + Q_A + Q_F$, kip-in.	+ 239	+ 287	+ 89	− 447	− 1377
29	$Q_H - Q_A - Q_F$, kip-in.	− 67	− 295	− 735	− 1487	− 2599

APPENDIX B

CORRECTION FOR TAPER RATIO

GIVEN: A control surface has square tips and a taper ratio $\lambda = 0.78$. The following characteristics for $\lambda = 0.45$ have been obtained from TMB Report 933:

Rudder angle, deg	7	14	21	28	35
Attack angle α, deg	5	10	15	20	25
Effective aspect ratio a_e	1.72	1.54	1.36	1.18	1.00
Lift coefficient C_L	0.193	0.373	0.534	0.673	0.787
Drag coefficient C_D	0.011	0.041	0.083	0.146	0.230
Normal force coefficient C_N	0.193	0.373	0.538	0.683	0.810
Resultant force coefficient C_R	0.193	0.373	0.539	0.686	0.815
Chordwise center of pressure (aft of leading edge of mean chord) CP_{LE}	0.175	0.189	0.209	0.235	0.266

TO FIND: Equivalent values for $\lambda = 0.78$.

PROCEDURE: For convenience, use subscript 1 for the given values ($\lambda_1 = 0.45$) and subscript 2 for the desired values ($\lambda_2 = 0.78$). Referring to Figure 28 of TMB 933, the crossflow drag coefficients are:

$$(C_{D_c})_2 = 1.335$$

$$(C_{D_c})_1 = 0.800$$

$$\Delta(C_{D_c}) = (C_{D_c})_2 - (C_{D_c})_1 = 0.535$$

From Equation [1] of TMB 933 we obtain:

$$\Delta C_L = (C_L)_2 - (C_L)_1 = \frac{(\Delta C_{D_c})(\alpha_r)^2}{a_e}$$

where α_r is the attack angle in radians. This is used in Line 6 of the detailed calculation sheet.

From Equation [2] of TMB 933 we obtain:

$$\Delta C_D = \frac{(C_L)_2^{\,2} - (C_L)_1^{\,2}}{2.83\ a_e}$$

This is used in Line 12 of the detailed calculation sheet.

From Equation [4] of TMB 933 we obtain:

$$\Delta(C_{m_{\frac{c}{4}}}) = -\frac{1}{2}\,\Delta C_L$$

This is used in Line 24 of the detailed calculation sheet.

The tabulation that follows is intended to be in a form that permits checking step-by-step. Certain operations are indicated by line number, for further clarification.

TABULATION OF CALCULATIONS FOR APPENDIX B

Line 1	Rudder angle, deg		7	14	21	28	35
2	Attack angle α, deg		5	10	15	20	25
3	Attack angle α_r		0.0873	0.1745	0.2618	0.3491	0.4363
4	$(\alpha_r)^2$		0.00762	0.0305	0.0685	0.1219	0.1904
5	a_e		1.72	1.54	1.36	1.18	1.00
6	$\Delta C_L = \dfrac{(0.535)(\alpha_r)^2}{a_e}$		0.002	0.011	0.27	0.055	0.102
7	$(C_L)_1$		0.193	0.373	0.534	0.673	0.787
8	$(C_L)_2 = (C_L)_1 + \Delta C_L$	6 + 7	0.195	0.384	0.561	0.728	0.889
9	$(C_L)_2^2$		0.03802	0.14746	0.31472	0.52998	0.79032
10	$(C_L)_1^2$		0.03725	0.13913	0.28516	0.45293	0.61937
11	$(C_L)_2^2 - (C_L)_1^2$	9 − 10	0.00077	0.00833	0.02956	0.07705	0.17095
12	$\Delta C_D = \dfrac{(C_L)_2^2 - (C_L)_1^2}{2.83\ a_e}$		0.0002	0.0019	0.0077	0.0231	0.0604
13	$(C_D)_1$		0.011	0.041	0.083	0.146	0.230
14	$(C_D)_2 = (C_D)_1 + \Delta C_D$	12 + 13	0.0112	0.0429	0.0907	0.1691	0.2904
15	$(C_N)_1$		0.193	0.373	0.538	0.663	0.810
16	$\cos\ \alpha$		0.9962	0.9848	0.9659	0.9397	0.9063
17	$\sin\ \alpha$		0.0872	0.1736	0.2588	0.3420	0.4226
18	$(C_L)_2 \cos\ \alpha$	8 × 16	0.1943	0.3782	0.5419	0.6841	0.8057
19	$(C_D)_2 \sin\ \alpha$	14 × 17	0.0010	0.0074	0.0235	0.0578	0.1227
20	$(C_N)_2$	18 + 19	0.195	0.386	0.565	0.742	0.928
21	$(CP_{LE})_1$		0.175	0.189	0.209	0.235	0.266
22	$(CP_{c/4})_1 = 0.25 - (CP_{LE})_1$	0.25 − 21	+0.075	+0.061	+0.041	+0.015	−0.016
23	$(C_{m_{c/4}})_1 = (CP_{c/4})_1\ (C_N)_1$	15 × 22	+0.0145	+0.0227	+0.0220	+0.0102	−0.0129
24	$\Delta(C_{m_{c/4}}) = -\dfrac{1}{2}\ \Delta C_L$	−0.5 × 6	−0.0010	−0.0055	−0.0135	−0.0275	−0.0510
25	$(C_{m_{c/4}})_2 = (C_{m_{c/4}})_1 + \Delta(C_{m_{c/4}})$	23 + 24	+0.0135	+0.0172	+0.0085	−0.0173	−0.0639
26	$(CP_{c/4})_2 = (C_{m_{c/4}})_2 \div (C_N)_2$	25 ÷ 20	+0.0692	+0.0446	+0.0150	−0.0233	−0.0688
27	$(CP_{LE})_2 = 0.25 - (CP_{c/4})_2$	0.25 − 26	+0.1808	+0.2054	+0.2350	+0.2733	+0.3188
28	$(C_D)_2^2$	14 × 14	0.00013	0.00184	0.00823	0.02859	0.08433
29	$(C_R)_2^2 = (C_L)_2^2 + (C_D)_2^2$	28 + 9	0.3815	0.14930	0.32295	0.55857	0.87465
30	$(C_R)_2$	$\sqrt{29}$	0.195	0.386	0.569	0.747	0.935

DISCUSSIONS

A. Suarez:

This paper gives some of the practical aspects of rudder design. We in the Laboratories haven't digested our information sufficiently so that a realistic rudder design can be made. Apparently, where our information has been consolidated sufficiently, those men who are responsible for actually putting a rudder on a vessel, so that the vessel will do what we think it should do, have sufficient technical knowledge. But in any event those rudders have to be designed. We have to contend with the practical engineer, and he still has to come out with a rudder engine, the strength of the shaft, the size of the rudder, so that we can still build vessels that can't wait.

Mr. Taplin is one of the designers who has to get an answer. He has been urging the Controllability Panel of SNAME to present some facts, to give him the tools, some information, so that we can build better and bigger ships. We still have, as declared by Lloyds of London in an article about two or three years ago, roughly a thousand ship accidents per year. Now we can't say that all of these accidents are associated with the handling qualities of the vessels. Some of them may be due to poor judgment, but of the thousand accidents, I believe, from what I have seen of many of the vessels which we have tested in the Davidson Laboratory, that we still have unstable vessels. This is due primarily to the fact that the operators insist that the construction cost of our vessels be kept at a minimum so that they can compete with foreign manufactured vessels and still stay in the market and still have an American fleet afloat.

J. L. Goldman:

I would like to make a comment as an engineer, and not as a designer, working with the ship operator. The importance of the rudder not failing is appreciated, and we are very conservative in our designs. The method outlined here considered the type of loading on the rudder which results when a ship enters a turn while proceeding full speed ahead. This might not necessarily be the maximum load that the rudder is going to sustain. For instance, in a seaway, wave slap under certain conditions might be important.

The ship operator has to be reasonably sure that the whole rudder system is not going to fail and we are always asked to be overconservative in the design of a rudder system and have a high factor of safety. We appreciate the importance of refining these calculations but the extra cost of adding a little more steel to the rudder system is usually requested by the shipowner who is usually also the ship operator.

F. S. Couldwell:

I have a very brief comment, or rather an addition to make to Mr. Taplin's paper which was very welcome to us. In the case of the DL1, we made a comparison of the results from tests that were reported in a DTMB paper in Transactions, SNAME, 1958.* Using this particular method, we got very good correlations when we used the 5:7 ratio of angle of attack to rudder deflection angle. Also by taking a speed reduction of around 28.0 percent, we were able to match the DL1 curves almost exactly.

S. Bindel:

In the case of a twin-screw ship with two rudders, do you fit the rudders just behind the propellers or off-center?

Mr. Taplin

One of the other things covered in Captain Saunders' 1944 report (I think it is) is that we make a big mistake if we put the rudder directly in line with the propeller shaft because we get a cavitation cone coming off the hub. For two reasons we keep the rudder off the shafting line: one is to avoid this cavitation cone, and the other is that sometimes it simplifies unshipping the shafting. But we would almost never put the rudder behind the propeller centerline.

C. R. Olson:

Mr. Taplin has not mentioned in his fine paper anything about model tests for determining rudder forces. Evidently, he does not have too much faith in our ability to correlate model results with full-scale results. Mr. Taplin uses aerodynamic test data for his calculations. We have good agreement in our open-water rudder force tests with aerodynamic test results. With our new maneuvering basin facilities, we should endeavor to find a hydrodynamic answer to this problem. If we do not have correlation between model and full-scale ship forces, then I believe we should still run the model tests to determine what the discrepancies are.

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*Becker, L.A. and Brock, J.S., "The Experimental Determination of Rudder Forces During Trials of USS NORFOLK," Transactions, The Society of Naval Architects and Marine Engineers, Vol. 66 (1958).

SHIP MANEUVERABILITY AS INFLUENCED BY THE TRANSIENT RESPONSE TO THE HELM

by

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ABSTRACT

The hypothesis is advanced that the maneuverability of a ship-helmsman combination is significantly influenced by the handling qualities of the vessel, these handling qualities being, in turn, a critical function of the transient response to the helm.

The objective of the presentation is to develop a fundamental understanding of helmsman/ship/steering-system behavior as it is related to the basic hydrodynamic design of the vessel and the characteristics of the human controller. Accordingly, a careful distinction is made between ship maneuverability and ship handling qualities before proceeding to review briefly what handling-qualities research with other vehicles has revealed about the nature of the human operator. This knowledge is applied to the particular problem of ship-control operations, and the conclusion is drawn that automatic-control systems will play an increasingly important role in improving the handling qualities of ships, provided appropriate research is performed to specify valid handling-qualities criteria for waterborne craft. The manner in which automatic control will assist the helmsman in extracting the full maneuver capability out of his vessel is discussed, and attention is directed toward the future, wherein ships will possess integrated directional- and seakeeping-control systems.

INTRODUCTION

To state that the design of a rudder and skeg for a ship is more of an art than a science is to repeat what has often been said before. Today, it is generally conceded that much remains to be done in order to ensure good ship maneuverability by means of rational design procedures. Thus, in recent years ship qualities such as course stability, ease of steering, and precise path-keeping and path-changing performance have been receiving increased attention from the scientific and technical community concerned with waterborne craft. In view of the increased attention to these matters—this symposium being a noteworthy example—it is believed that a good purpose would be served by examining the problem of ship maneuverability from the *systems* viewpoint. Such a review would examine ship maneuverability as it is influenced and determined by the characteristics of:

1. the ship
2. the control system, and
3. the helmsman.

It is hoped that this kind of review will tell us where we stand today with perhaps a new and unorthodox viewpoint (perhaps also controversial), and in so doing, will assist in the formulation of research programs required to convert existing design procedures from an art to precise technology.

Today, when limited knowledge or theory exists, the engineer or designer very frequently relies on past experience for direction and guidance. Thus, ships characterized by operators as having satisfactory steering and maneuvering qualities often serve as a guide for future designs. In those cases where the designer is in doubt, he can resort to tank tests in order to check the behavior of his latest creation. But how does the designer proceed to advance the state of the art? Since refinements in design often lead to technical complexities and perhaps additional costs, what criteria can the designer use in making a choice between several design alternatives or in selecting a final compromise? Questions that are particularly basic to the specific topic under discussion are:

> How maneuverable should ships be in order to perform their designated missions?
>
> How easy should they be to control?
>
> What are the benefits, both from the considerations of economics and safety, if the helmsman can perform his task with greater ease and precision, or if ship control were a task made sufficiently simple that any member of the crew could take over the helm?

These questions must, at present, go unanswered. Yet it appears that answers should be obtained in order to determine the degree to which advances—substantial or otherwise—are required in the stability and control of waterborne craft. The argument is advanced that valid answers will be obtained only by approaching the problem from a systems or integrated viewpoint. This paper does not propose to examine or answer all the questions raised above. Rather, it will strive to develop a fundamental understanding of helmsman/steering-gear/ship-system behavior as it bears on the design conflicts inherent in achieving good maneuverability and good handling qualities. It is hoped that these efforts will permit valid conclusions to be drawn on the role that automatic-control systems can and should play in improving the handling qualities of ships and thereby aid the helmsman in extracting the full maneuver capability out of his vessel. Another objective of the paper is to postulate a theory of man/control system/ship behavior that will pinpoint the kind of research needed to answer some of the basic questions raised above.

A REDEFINITION OF TERMS

No attempt will be made here to review all of the literature pertinent to the problems of steering and maneuvering waterborne craft. It is sufficient to note that ship builders, designers, and operators have been concerned with this problem from perhaps the earliest days of water transportation up to the present time, when technologists are devoting an

exponentially increasing amount of attention to ship stability and control.

The recent (i.e., starting in the early forties) and current treatments of ship stability and control have dealt with many aspects of this general topic and have succeeded in formalizing, to a degree, suitable and appropriate terminology. Thus we have *turning*, *maneuvering*, and *course changing* or *path changing* as synonymous concepts which embrace both the steady-state and transient characteristics of the ship/steering-system combination. Similarly, *steering* and *course keeping* or *path keeping* are generally accepted as synonymous concepts that also embrace static and dynamic behavior of the elements involved.

Traditionally, a majority of authors have considered only the hydrodynamic characteristics of the hull and its appendages as having bearing on the course-keeping and/or course-changing properties of a vessel. In contrast, Dieudonné[1] has stressed that course stability (stability of route) is significantly influenced also by the performance of the steering gear, namely, its static resolution and dynamic behavior (i.e., followup lag between helm and rudder). In his words:

> "Maneuverability is the readiness or the ability of a ship when traveling in good weather and in a calm sea, to take the path which the steersman desires it to follow. This depends upon:
>
> 1. The rapidity of the response of the ship to the action of the rudder. This in turn is influenced by the rapidity of shifting the helm and rudders, that is to say, by the characteristic of the steering apparatus."

Note that Dieudonné defines maneuverability as a *ship* property which, in turn, is a function of both the turning response to rudder and the response of the steering apparatus. In addition, he brings the human operator into his definition by referring to "the path which the steersman desires to follow." A definition of this type raises the question as to what is meant by the concepts of turning and course-keeping qualities. Are we referring only to ship behavior, as it influences these concepts, or do we mean the behavior of a closed-loop system where the human controller is an essential part of the loop? In the past, this distinction has not been drawn very carefully. This is not surprising since it is indeed difficult to define rigorously the handling-qualities terms that properly account for the human element in the system. This is because of the tendency for objective measures of performance to become inextricably intermingled with criteria of performance.

The necessity of adopting a *systems* viewpoint is thereby emphasized, and it is most important that terminology be carefully defined in order to proceed to examine ship maneuverability from a systems point of view. For example, the terms *course changing* (path changing) and *course keeping* (path keeping) have been mentioned above, without definition, as two concepts that have relevance to the problem of positioning and orienting a vessel in the horizontal plane. We can sharpen up the definition of these two terms by, first of all, stating that they describe two separate aspects of the handling qualities of a vehicle moving

[1]References are listed on page 164.

in a horizontal plane. Second, we must note that handling qualities are subjective in nature; that is, the term implies an evaluation by the helmsman of the following two properties of a waterborne craft:

1. the ease and precision with which it is possible to turn the ship or achieve a desired path, and

2. the ease and precision with which a path or heading is maintained.

The first subjective property is denoted by the term *course-changing* qualities, and the second, by *course-keeping* qualities.

It should be pointed out, however, that these two *subjective* propertie. ·an be related to *objective* measures of ship-maneuvering and course-keeping behavior. Thus, it is a well established fact that the objective problem of measuring and/or defining the maneuvering performance of a ship can be solved by evaluating the lateral stability and control characteristics of a ship and its steering system. Further, objective measures of performance can be stated in quantitative terms, independent of any subjective evaluation or application of criteria derived from closed loop performance.

Within the framework provided by the above discussion, the handling-qualities problem is defined as follows:

> The determination of satisfactory handling qualities constitutes, in substance, the isolation of those objective measures of ship and steering-gear performance which, when presented to a human controller in the process of closing the control loop, produces a satisfactory subjective opinion and/or satisfactory performance of the man-machine combination.

In summary, there are objective measures of directional performance (stability and control) which can be transformed to a measure of handling qualities when the helmsman is brought into consideration through his closure of the control loop. Stability and control properties and steering-gear properties which produce good handling qualities, (that is, good closed-loop performance) can therefore be related to good maneuverability when maneuverability is defined as a performance characteristic of a helmsman/steering-gear/ship system.

THE ROLE OF THE HUMAN OPERATOR

A purely heuristic reasoning process leads one to conclude that it is the characteristics or capabilities of the human controller which determine whether a given set of stability and control characteristics will produce satisfactory closed-loop performance. If a block diagram of the helmsman/steering-gear/ship system is drawn (Figure 1), wherein the human operator is treated as a servo-system element, the performance of this system can be analyzed, provided the characteristics of each component in the system are known. Placing the problem in this form gives us a more logically compelling method of defining handling qualities by stating that specific steering-gear and ship dynamic characteristics are conducive

to good handling qualities when adequate closed-loop performance is achieved with minimum burdens being placed on the dynamic performance of the helmsman.

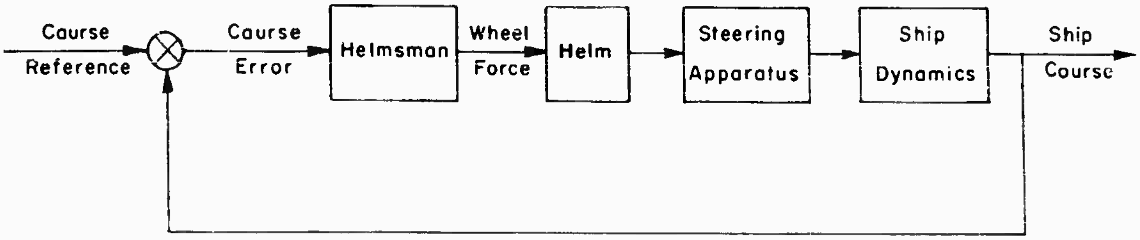

Figure 1 – The Helmsman/Steering Gear/Ship System

The system approach represented by Figure 1 and the above advanced hypothesis on handling qualities have been basic to the efforts now being expended in the aeronautical field to advance the state of the art in:

1. improving aircraft flying qualities,
2. defining valid handling qualities criteria for aircraft, and
3. improving closed-loop performance of tracking systems that include a human operator.

To this end, a significant amount of research has been performed following two general approaches. One approach has been to systematically gather pilot-opinion data as a means of bounding the stability and control characteristics that are conducive to good handling qualities, and the other approach has been to develop an understanding of the human operator as a servo element such that feedback-control system techniques could be used to synthesize an optimum closed-loop system.

Although this work will not be reviewed in detail, certain results obtained from research performed in the aeronautical field and from stability and control and handling-qualities investigations performed with other vehicles will be summarized, since this research, in the aggregate, has produced a qualitative understanding of man and the role he plays in vehicle control. It is believed that a review of this knowledge will be of advantage in placing into proper perspective subsequent discussion of helmsman/steering-gear/ship system performance.

The Air Force-sponsored studies have shown that the inherent adaptability of man in a system makes it impossible to formulate a general "human transfer function" even for simple tracking experiments performed in the laboratory. Certainly the dynamic model of man as developed by McRuer and Krendel[2] is based on laboratory experimentation that is far removed from reality as experienced by man in his attempts to control and guide a vehicle. Nevertheless, these tentative models have led to a conclusion derived by servo-engineering procedures that agree with a similar conclusion derived through psychological reasoning.[3] This conclusion is that optimum system performance will be obtained when the "least demands" are made on the human operator, namely man becomes the equivalent of a simple amplifier.

This is equivalent to stating that the transfer characteristic of the controlled element should possess approximately zero dynamics.

It should not be overlooked that these results have been derived from analyses of restricted tracking experiments that are, as was previously mentioned, situations that are far removed from real-life control of vehicles. Nevertheless, it has been possible to use a human-servo model in conjunction with known aircraft dynamic characteristics to perform an analysis that correlates remarkably well with pilot opinion data obtained in flight tests. These findings point up the validity of the following concept, as expressed by Westbrook and McRuer:[4] "pilot opinion of an airframe configuration is closely correlated with closed-loop performance, and hence to some extent with the transfer characteristics and parameters adapted by the pilot to control the configuration." Again the experimental data support the conclusion that optimum system performance is obtained and therefore, the controlled element possesses optimum handling qualities, when the human controller is permitted considerable variation in his transfer function. This allowable variation in performance of the human controller—in other words, no rigid demands are placed on his transfer function—is a result that bears considerable resemblance to the "least demands" theory postulated by Birmingham and Taylor and substantiated, to a remarkable degree, by McRuer and Krendel.[5]

To recapitulate, the tracking experiments indicate that the dynamics of the controlled element should be practically nonexistent to produce optimum closed-loop performance. This result can be interpreted in either of two ways:

1. Zero dynamics between the output of man (i.e., control force or displacement) and the output of the controlled element means that no integration or differentiation is required on the part of the human operator and he can thus perform his control task in a manner analogous to a simple amplifier.

2. Zero dynamics means that there are essentially no lags or time delays between his output and the vehicle response he aspires to produce.

The question then arises: How well does the second interpretation agree with the results of flight test and our everyday experience in the control of other vehicles?

Data obtained in research programs examining the longitudinal flying qualities of aircraft[6] result in the iso-opinion chart pictured in Figure 2. It is seen that the short period natural frequencies and damping ratios associated with good handling qualities, as measured by pilot opinion, represent, on the average, a second-order dynamic system having a response time of 1.0 second, where response time is defined as the time required for the transient response to reach and remain within 95 percent of its steady-state value. This is a result which, although strictly applicable only to the performance of longitudinal flying tasks in a fighter aircraft, demonstrates that the second interpretation (of the real significance of the requirement for zero dynamics) has validity. The hypothesis is substantiated by our everyday experience in the control of automobiles where the existence of good handling qualities is indicated by the very small learning time required for the average person to learn the steering process

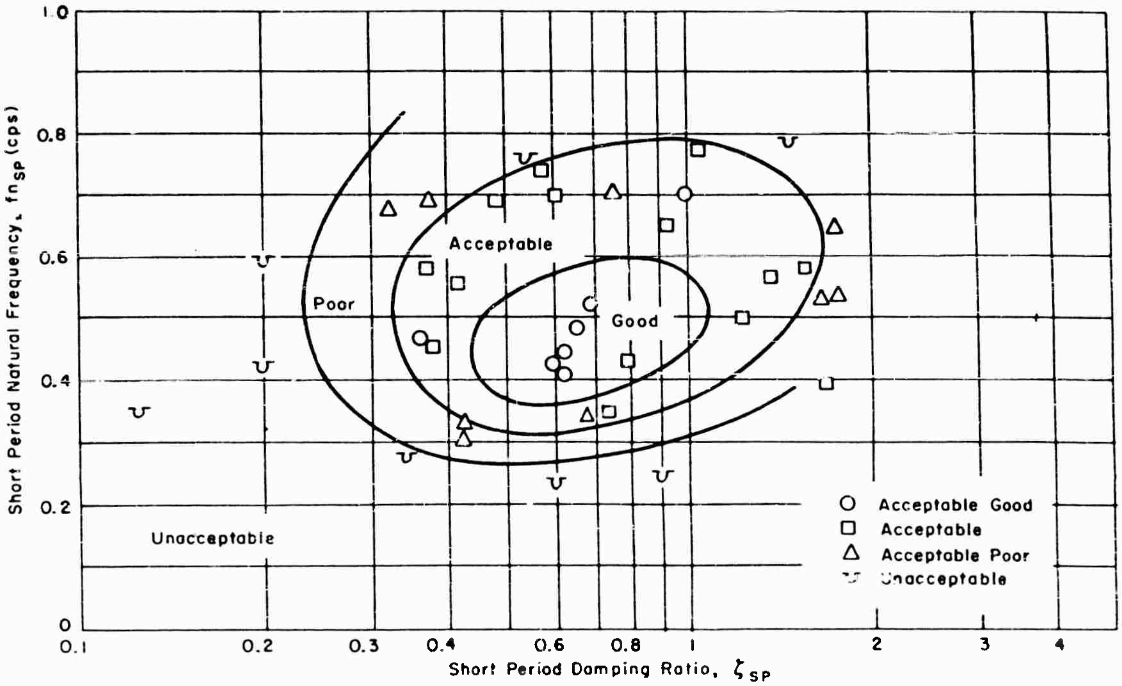

Figure 2 – Aircraft Longitudinal Handling Qualities Criteria as Derived from Pilot Opinion Data

and furthermore execute very precise path-following and path-changing maneuvers. Figure 3 shows that typical automobiles have essentially instantaneous response at very low speeds with the response time increasing to approximately 1 second for a directionally stable automobile moving at 60 miles per hour. It is seen that these response times are akin to those exhibited by a typical subsonic fighter aircraft.

On the other hand, there are examples of slower responding vehicles that are known to be difficult to control and that have been universally adjudged to possess poor handling qualities. The helicopter is an excellent example. The response time is on the order of 10.0 seconds in certain flight configurations wherein it is dynamically stable or has been made stable by appropriate means. It has been observed[7] that this long response time in helicopters is responsible for a control behavior mode on the part of man that is best termed ***overcontrolling***. This behavior can also be observed during the manual control of surface vessels and submarines; however, full discussion of this point will be made later in the paper.

Thus far, an attempt has been made to show that a substantial body of research evidence demonstrates the human controller to be so constituted such that optimum closed-loop performance of a man/control/vehicle system is attained only when adequate restrictions are placed on the dynamics of the vehicle/control-system combination. These restrictions have been demonstrated elsewhere to be a dual function of:

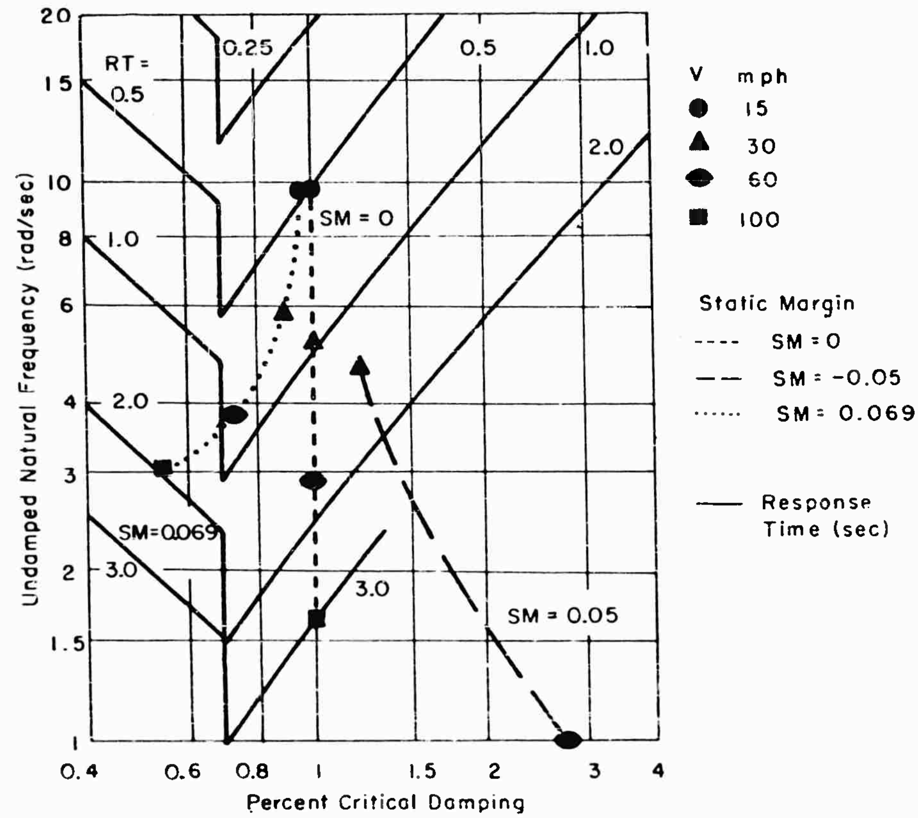

Figure 3 – Frequency and Damping Characteristics of the Directional Mode of Automobiles (for Three Levels of Static Margin)

1. man's dynamic behavior, and

2. the control task specified by the *mission* of the vehicle and the *environment* in which it operates.

Although major stress has been placed on the dynamic characteristics of the vehicle/control system in determining handling qualities, the influence of the various steady-state characteristics of the vehicle/control system should not be overlooked in this regard. For example, the static sensitivities, as defined by the ratio of vehicle steady-state response to control displacement and/or control force, have been found[6] to play a very important role in the definition of handling qualities. Similarly, laboratory studies have shown[8] that the presence of friction, backlash, and flexibility in a control system significantly influence aircraft handling qualities in pursuit-tracking experiments. Full-scale experiments with automobiles have also demonstrated the importance of these control parameters in influencing overall *control quality*.

THE DYNAMICS OF WATERBORNE CRAFT CONTROLLED BY HUMAN OPERATORS

Unfortunately, one cannot apply the previously discussed concepts to ships as conveniently as to other vehicles since waterborne craft tend to be significantly nonlinear in their behavior. The characteristic nonlinear relationship between ship-rudder angle and steady-state yawing velocity is due, of course, to the nonlinear hydrodynamic characteristics of the hull and the nonlinear control effectiveness of the rudder. Furthermore, the handling-qualities discussion in dealing with response times, assumed, *a priori*, that the vehicle/control system was stable. It is a matter of common experience that certain ships exhibit dynamic instability for small ranges of turning velocity and become stable in turning after a certain level of path curvature has been reached. Full-scale experience has demonstrated[1,9] that such behavior is not conducive to good course keeping and therefore is detrimental to the overall achievement of good handling qualities.

Notwithstanding the tendency of waterborne craft to exhibit nonlinear hydrodynamic characteristics, it is still proper to consider their dynamic-response characteristics irrespective of the degree to which this response is influenced by the nonlinear force and moment characteristics of the hull and its appendages. It is immediately observed that the response times that are characteristic of ships place these craft into an operating realm that is completely divorced from the time scale that is operative with vehicles such as aircraft and automobiles. The question now arises: To what degree can we utilize our knowledge about human behavior as it determines the handling qualities of fast-responding vehicles to extend our understanding of ship-handling qualities and ship maneuverability?

Before proceeding to discuss this question and its possible answers, certain observations can and should be made. First of all, even a casual observation of the manner in which a helmsman and a driver of an automobile both perform their steering tasks shows that there is a considerable difference in their behavior. The latter rarely uses all of the steering control that he has available (except when parking, making U turns, etc.) and by and large, acts as a proportional controller, turning the wheel in proportion to the steady-state path curvature that is required to follow the road. The helmsman, however, can be observed to make course changes by swinging the helm hard over and then checking his turn by swinging his helm hard in the other direction. This is followed by a return of the helm to neutral; that is, if he is skilled and intimately knows his ship; otherwise several oscillations of the helm will occur before he steadies down on his desired course. It appears that the helmsman is acting like a "bang-bang" controller and, in truth, this "bang-bang" control principle is analogous to what is taking place.

Earlier, mention was made of the tendency for operators to *overcontrol* slow-responding systems. This *overcontrolling* can easily be observed with unskilled operators attempting to maneuver a ship or a submarine. It is now pertinent to ask whether the model of the human controller discussed earlier (specifically his desire to control a system with minimum

dynamics, i.e., minimum time lags) explains this tendency to overcontrol or to operate in a "bang-bang" manner. A servo analysis will immediately show that, if the human operator is not content with controlling a slow system and desires to speed up the response (a concept called *response augmentation*), the "bang-bang" control mode will automatically appear. This result will occur particularly in those instances where the controlled element has limited control power (ratio of available control moment to inertia) as is the case for ships.

It now becomes instructive to examine the closed-loop behavior exhibited by a helmsman/steering-gear/ship system with the aid of purely intuitive reasoning. On the open sea his threshold for visual sensing of yaw rate or path curvature is appreciably higher than that of the driver of a car. Presumably, this helmsman, with human limitations on (1) perception and (2) ability to integrate into the future, is not willing to set a rudder angle that will eventually produce the desired rate of yaw but which, for a significant duration subsequent to his action at the helm, will produce an imperceptible response. Accordingly, he puts the helm over hard in such manner that the resulting response is sufficient to establish that he has control over his ship.

This line of reasoning could be continued to explain the remainder of the process. However, it is evident that the process described above will produce the same saturation type of control that results from a servo-oriented analysis procedure.

On the basis of the foregoing discussion, it is submitted that:

1. the knowledge obtained from human-dynamics research,
2. the results derived from pure intuitive reasoning, and
3. everyday experience and observation

all support the general conclusion that a helmsman will attempt to augment (i.e., speed up) the response of his ship as he proceeds to carry out his control task. It is further postulated that the degree to which a human controller attempts to speed up a slow responding system is, in addition to being a function of the transfer characteristics of man, a critical function of the operational task; namely, the required precision of steering.

This latter point (namely, the required precision of steering) is a concept that is fundamental to an integrated assessment of ship maneuverability. As indicated by Saunders,[10] this is one of the areas in which knowledge is so woefully lacking. It would appear that a program of research, properly conceived and performed with suitably modified ships, will be essential in order to establish valid handling qualities criteria as a function of the maneuvers a ship is typically required to perform. (Needless to say, it is conceivable that a higher order of handling qualities could make possible closed-loop maneuvers that under present circumstances are never attempted by experienced helmsmen.)

Although it is probably universally recognized by naval technologists that there is some minimum response level which designers should provide in order to establish adequate turning and course-keeping qualities, there is probably no consensus of opinion on what this minimum level should be. In 1946, Davidson and Schiff[11] proposed a set of hydrodynamic

design goals that would, in effect, provide a ship response to rudder believed by these investigators to ensure good turning and course-keeping qualities. Their conclusions were based on an examination of existing design practice and bore no relationship whatsoever to the basic requirements of the helmsman closing the loop or to any quantitative measure of required steering performance or precision. Their analysis was an epochal event in the field of ship stability and control, but subsequent investigators have not appreciably advanced the state of the art other than to belabor the point that quickened response or improved stability is attained at the expense of maximum steady-state turning performance. It is precisely this compromise that is at the heart of the entire problem, and it is argued here that every effort should be exerted to acquire the research data that are needed to assist the designer in resolving his dilemma.

THE ROLE OF AUTOMATIC CONTROL IN THE IMPROVEMENT OF SHIP MANEUVERABILITY

In view of the lack of handling-qualities criteria for slow responding systems such as ships (that is, criteria obtained from research that accounts for human-operator performance and steering-task requirements), the following hypothesis is advanced:

> A suitable interim measure of handling qualities for ships is the degree to which a helmsman provides response augmentation in performing typical maneuvers of course-changing, path-keeping, etc. Control actions containing a large degree of response augmentation indicate poor handling qualities, and control actions containing little or no response augmentation indicate good handling qualities and therefore good closed-loop steering performance.

The above statement is based on the hypothesis that any and all steering inputs by the helmsman in excess of what would be supplied when he is permitted to act as a proportional controller represents demands on his dynamic-transfer capabilities that will in all likelihood:

1. reduce his subjective rating of the ship,
2. increase the training time necessary to perform the required steering tasks with the desirable degree of precision,
3. result in less than optimum closed-loop steering performance if the operator is unskilled, and
4. prevent even the skilled helmsman from performing precise maneuvers under very unfavorable operating conditions.

It is quite probable that the proposed interim standard for good handling qualities will require ship stability and control characteristics that cannot be achieved by suitable hydrodynamic design. Ships, by virtue of all the design considerations that must be brought to bear, invariably possess large yawing moments of inertia relative to their dynamic stability or their total hydrodynamic stiffness in turning. A corollary feature of this small resistance to turning is that they need relatively small rudders to produce steady-state turning radii that

are exceptionally small when viewed on a nondimensional basis. Manifestly, small control powers and very low levels of dynamic stability are not conducive to producing a fast-respond-responding system. It appears that the only means by which significant gains could be achieved in shortening the response times of large waterborne craft would be to resort to automatic-control systems that position the rudder as a function of the response of the ship.

Ample descriptions of the principles and design details of response augmentation systems can be found in the aeronautical literature. Recently a study,[12] performed to (1) establish the information requirements for submarine control and (2) determine the optimum location of the human operator in the ship-control loop, resulted in the application of a response augmentation system to yield the optimum trajectories (limit maneuvers) that could be executed with a specific hull. Although the concept of a limit maneuver was not new, the ultimate maneuvering performance of the submarine was defined in a unique way. Specifically, a feedback-control-system analysis was performed to yield a closed-loop system, whose response was as rapid and well damped as is possible to achieve with the given hull and hydrodynamic control surfaces. This study demonstrated that the handling qualities of the conventional submarine are such that the full maneuver capabilities of the submarine cannot be exploited by the human controller. The provision of response augmentation through automatic control systems would undoubtedly be a step in the direction whereby the manual controller is given increased ability to perform more precise and more rapid maneuvers.

It should be noted that response augmenters, by their very nature, are stability augmenters as well. On application, they also eliminate the hysteresis loop observed in the so-called spiral maneuver and thus markedly improve the course-keeping qualities of an otherwise unstable ship.

The above remarks on the role of response augmentation indicate that the time may not be too far distant when feedback-control systems will be utilized to improve the handling qualities of seagoing vessels, as well as their seakeeping properties. Response augmentation systems should not be confused with autopilot-type systems that provide ships with a sensitivity to an ordered heading or provide a submarine with a sensitivity to an ordered depth. Rather, we are speaking of systems that augment or alter the dynamics of a ship, making it more amenable to precise manual control. Recognition should also be given to the fact that augmentation of the turning response could conceivably increase the roll excitation due to turning. In such an event, the roll dynamics of the ship would couple to an increased extent with its turning dynamics. If this proves to be the case, consideration would conceivably be given to coupling the roll stabilizers of a ship with the control system actuating the rudder.

Irrespective of whether the response-augmentation concept will ultimately be exploited in future ship designs, it will necessarily play an important role in conducting research experiments required to shed light on the overall ship handling-qualities problem. The need for this research was stressed earlier in the presentation, and it is encouraging to note that the proper performance of these necessary handling-qualities investigations are in no wise restricted by the present state of the control art.

CONCLUDING REMARKS

In summary, it has been postulated that:

1. Ship-response characteristics required for good handling qualities (i.e., good man-machine performance and high rating of response by subjective opinion) are, in part, a function of the transfer characteristic of man and, in part, a function of the operational task; namely, the required steering maneuvers and the required precision of execution.

2. Valid handling-qualities criteria for surface ships can only be defined by carefully conducted experiments with full-scale ships, in which proper recognition is given to each element of the closed-loop system and proper distinction is made between static and dynamic phenomena.

3. Lacking valid, substantiated criteria for assessing handling qualities of vehicles possessing extremely long response times, a suitable interim measure of handling qualities is the degree to which a human controller provides response augmentation in performing typical maneuvers, such as course changing, path keeping, etc.

4. The nature of ship hydrodynamics is such that drastically improved response is not likely to be achieved through hydrodynamic design of the hull, but rather by means of increased rudder power actuated by an automatic control system.

5. The transient response to the helm is fundamental to the determination of handling qualities, which, in turn, govern the speed and preciseness of control (i.e., maneuverability) that can be exhibited by a helmsman/steering-gear/ship system.

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DISCUSSIONS

H.E. Saunders:

I am going to have to be a little careful of what I say now. I don't want to detract from the valiant efforts of the author or from the results which have been achieved in the conclusion, but I had hoped that the author of this paper might tell us the story of it in words that we can understand. I am afraid that a great deal of the meaning of this paper which I read at home, in quiet and leisure is lost upon me because I just didn't know what you were trying to say. It is difficult of course when someone comes out of a select circle, as it were, and uses his specialized language for the benefit of other people. Many of you, I am sure, are facing what I am facing now, to find our way through what they might call the statistical phase for naval architects. Now I'm a little bit handicapped because I'm very much interested in what the author is trying to do and what he is trying to say, but I am going to have to go home and translate this or get it translated.

However, there is one point which I think I gathered well enough to comment upon. I really believe that the author here has missed one whole factor in his box diagram and loop. And that factor is the person or whatever agency which gives the orders. Mr. Bindel's paper, which will be given this afternoon, makes continual mention of the pilot who is directing the operations in the model basin under the carriage. Those of you who were here in 1946, 1947, and 1948 will remember that when we did the modernization job for the Panama Canal Authority, we had successively Panama Canal pilots who came up here for a month, who rode the carriage on every ride and actually gave orders. The orders were not given by the TMB staff. The orders were not automatic; they were given by the pilot. The steersman operated the controls and endeavored to do his best to carry out the pilot's orders. Mr. Bindel's paper also mentions the pilot.

Now then, one might say, how about running in the open sea such as in the diagram which you show? You begin with a course reference. It so happens that with present gyro compasses, you get a pretty good course reference. Actually, you get a heading reference because you don't know what the course is. But in the old days, for those of you who are old enough to have steered ships by compasses, you had no really good heading reference, so you had to anticipate the magnetic compass; there you had another factor.

When you come to emergency maneuvers, such as piloting, handling ships around docks (or even trying to catch turtles), whatever you do, the steersman gets his orders from other people. The steersman is rarely the piloting officer. It so happens that in our icebreakers, the piloting officer, the ice pilot, is the man who operates the rudder controls, so he doesn't have to give orders to anybody; he doesn't have to say a word. But in all other cases there must be a pilot; there must be a commanding officer, a captain, first mate, or someone, who gives the orders. So you can't eliminate this one more factor.

As Mr. Bindel brings out in his paper, for a model test with a pilot or with separate director, he has to work fast because the times are reduced by the square root of the linear ratio of full-scale to model size.

I hope that maybe the author will find time to give another drawing or another diagram in place of Figure 3. I have been operating automobiles for 46 years now and yet I don't get anything out of the figure that was presented.

S. M. Y. Lum:

There is no disagreement with the author's persistent theme that more sophisticated studies of the control loop, using feedback techniques incorporating the helmsman, can improve the handling qualities and response of ships. However, with due respect to Mr. Segel, I wonder if he is not putting undue emphasis on the control loop as the panacea for all design ineptness and at this stage, placing the "cart before the horse," so to speak. I would say there is an area which takes precedence to that and which could stand more intensive study and clarification—and that is in the open-loop characteristics of the ship. With due apologies to Captain Saunders (whose preceding discussion on the paper assailed the author's use of control semantics making it extremely difficult for this type of audience to follow), by the open-loop problem, I refer to the need for more exact delineation of the ship's transfer function which relates the hydrodynamic derivatives in terms of ship geometry and arrangement. That is, if the hydrodynamic derivatives for a complex shape such as that of a surface ship operating on the air-water interface can readily be determined, then it would be opportune to get the maximum benefits from a more advanced and sophisticated closed-loop study as a natural followup.

The nature of the initial difficulty, in the case of surface ship, for the one part, lies in pinpointing the open-loop transfer function with all the usual associated problems of nonlinearities, cross-coupling, free surface, etc. The other part lies in attaching a meaningful input to the system in view of the irregularity of the sea with the associated problems in the event of nonstationary processes, non-Gaussian distributions, etc.

When such information as above can be jelled into a pole-zero configuration in the complex plane, early decisions can be made. If the hydrodynamic design is found to be excessively unstable (open-loop-wise), or to have large response amplitudes within the expected frequency band of operation, this will be reflected in these poles and zeroes. For such bad preliminary designs, it would not be good practice to "close the loop" by brute force at the expense of having an "exotic" automatic control loop. It would be better to modify the design in the early stages of the development instead of discovering that there is a bad point of instability after the ship is built. Then to redeem the ship at this stage with fancy controls would be like chasing a failing investment with more good money.

Finally, before turning over the open loop to the control people to "close" with all the automatic sophistication as may be judiciously required, it is essential to ask the question: What is the functional or mission requirement against which all other compromises should be weighed?

Going back to the aircraft concept of designing for a slow-speed cargo vehicle, maneuverability may be relaxed in favor of course keeping. On the other hand, an attack vessel should be favored with high maneuverability. The closure of the loop could then, in the latter case, incorporate all the sophistication of stability and response augmentation.

In closing, I do not wish to sound discouraging to Mr. Segel. In fact, I believe that with the advent of more sophisticated naval craft such as the hydrofoil boats and GEM craft, there will be a great need for this specialty. However, everything should be considered in due sequence and due perspective. Thank you for an interesting injection into the field of ship maneuverability.

V. E. Williams:

First of all, I would like to congratulate Mr. Segel on his paper. I think it was well done. Unlike Captain Saunders, I am familiar with both jargons but I have the same problem at times. I would like to mention, though, that I believe we are right on the threshold of all these things that Mr. Segel is talking about, and like the former discusser here, I also believe we have to prepare ourselves for this work. We (at Sperry) have found that we had to get the dynamics of the boat. It wasn't enough just to use the linear yaw equations. Having defined very well all of the disturbances in this test, the seaway, etc., we had to define our inputs. And, as Captain Saunders said, that means the captain and from the captain to the helmsman, and in that context we find what orders would come from both and how they would react. I believe the marine field is on the threshold of doing this now. I know that Sperry itself has worked on many of these things and have some very good mathematical models of ships, and particularly submarines. With the aid of the Stability and Control Division of the Model Basin we now have a nonlinear model, for example, where we can get all the turning maneuvers, and can duplicate the Dieudonné spirals. Also with the help of the oceanographers at the Model Basin, we have the spectrum of the seas which are linear but very complex, and with the aid of the Seaworthiness Division of the Model Basin, we can judge the forces and moments due to waves on the ship. Now it is a matter of putting all this together and getting to work.

A. Goodman:

I have a brief comment, but before I go into it I would like to make one remark while the paper is fresh in everyone's mind. One item that was also eliminated in this closed-loop diagram which was not mentioned by the other two comments was the display system. This factor is very important; it is the source of information that the controller has, and its characteristics will influence the performance as well in the closed-loop circuit.

Handling qualities, as viewed by the Model Basin, are not subjective in nature, which is a point held by the author.* Definitive maneuvers are designed to evaluate the inherent as well as closed-loop characteristics of the given system. The paper given by Mr. Gertler and Mr. Gover clearly defines the concept, and outlines many of these definitive maneuvers and the numerical measures associated with each.

The present paper contains a remark that concerns using pilot opinion data as a means of determining performance characteristics. In this regard, the Model Basin several years ago made an extensive survey of ship operators using a standard questionnaire. This survey resulted in many conflicting and contradictory answers regarding the handling qualities of sister ships. We considered the results of the survey as adequate proof that there is need for obtaining objective measures of the various handling qualities of ships. This, of course, led to the development of the definitive maneuvers by the Taylor Model Basin.

The author made another interesting statement in regard to the compromise the designer has to face between good directional stability and good control response. For some years we at the Model Basin have felt that this compromise need not be made. Now, it has been demonstrated on vehicles such as the SKIPJACK that such compromises need not be made and that both excellent course-keeping and course-changing characteristics can be achieved simultaneously. Also it is believed that the use of nonconventional control surfaces will further eliminate the need for such compromises.

The author makes a statement to the effect that good handling qualities will require ship stability and control characteristics that cannot be achieved by suitable hydrodynamic design. The objective of the Model Basin in this area is to relate the geometric characteristics of the ship to the handling qualities. This is what the ship designer really wants.

The author states that the only means by which significant gains could be achieved in shortening the response time of large watergoing craft would be to resort to automatic control systems that position the rudder as a function of the response of the ship. I can't quite see this. It seems to me that the ship cannot exceed its inherent performance. For example, if you use the rudder to ease into a turn, I believe that the result is a larger tactical diameter and a longer time to change heading, etc. Some gain would probably be a smaller loss of speed in the turn and reduction of heel in turn.

I am a little confused by the statement "The provision of response augmentation through automatic control systems would undoubtedly be a step in the direction whereby the manual controller is given increased ability to perform more precise and more rapid maneuvers." How does a man enter a system that is under automatic control? In other words, to override it. I'm not quite clear on exactly what you mean by response augmentation. I wonder if you could amplify this and also indicate to me how response augmentation has increased the stability of the system.

In his concluding remarks, item 2, the author made a statement that "Valid handling-qualities criteria for surface ships can only be defined by carefully conducted experiments with full-scale ships . . . ." The Model Basin does not believe this to be correct. It has

shown that the handling qualities of ships can be determined accurately on the basis of free-running models and analog computer and simulator studies utilizing experimentally determined coefficients for the equations of motion. The main objective of the Model Basin in this area is to provide performance evaluation results and design recommendations for improvement of performance, if necessary, *before* the contract plans are signed. Full-scale studies are performed mainly for correlation purposes. In using an analog computer as Mr. Williams points out, you can simulate the various components that go into making up the closed-loop system. In fact, in some cases you can actually insert an operational component (hardware) into the system, and this is a very powerful tool which has been in use at the Model Basin for the past four or five years. The author's thesis, on the other hand, is an after-the-fact approach which would be prohibitively expensive in ship work.

Item 4, "The nature of ship hydrodynamics is such that drastically improved response is not likely to be achieved through hydrodynamic design of the hull, but rather by means of increased rudder power actuated by an automatic control system." How do you increase rudder power with an automatic control system? Possibly, by increasing rudder rate, which could result only in comparatively slight gains. And there you have to consider the limitations of your steering mechanism which is normally designed to move the rudder at a rate of a few degrees per second. Even if major changes in the design of such systems were conducted, it is difficult for me to see that there would be any marked improvement in handling qualities as the result. I would like your opinion.

Robert Morse:

The author stated that a human operator would perform best when the least signal "shaping" is demanded of him; i.e., that he perform as a simple amplifier. It is pointed out that this is actually a very strict and demanding requirement, since a human being has an inherent lag in his response which, however, may be negligible when considering the long response time of ships. But it is certainly not negligible for the case of airplanes and automobiles as mentioned in illustration by the author. It is further pointed out that a certain amount of anticipation or "rate" is required of a human operator at the helm of a ship or the steering wheel of a car for efficient performance. Thus, an experienced helmsman will apply rudder as a function of the rate of change of heading (determined by the rate of travel of the compass fine indicator, for example) rather than waiting for an actual heading change before taking corrective action.

It would probably be in order to define a "good helmsman" as one who, through experience and natural aptitude, has so developed his own "transfer function" that in combination with the ship and environment, the overall system characteristics are near-optimum. To do this he must perform all of the functions of the automatic controller: that of sensing the error with his eyes or by "feel"; that of shaping and amplifying the signal in his brain by his experience and aptitude; and that of positioning the controls by his muscles and dexterity.

He has the advantage over most automatic controllers in that he is adaptive; that is, he is able to vary his "gain" and "signal-shaping" as the situation dictates. Thus, when he is at the helm in rough seas, he is more alert and takes action more quickly. In so doing he has actually increased his gain and phase lead (or anticipation) to perform the job. He can also change his criteria of performance at will. Thus, in rough seas he may yield on maintaining a tight heading control to limit excessive rudder activity.

The human operator has the disadvantage that his characteristics vary from minute to minute and certainly from operator to operator. His peak performance is limited by factors over which we have no control. He cannot match the automatic controller in sensitivity or consistency of action or constant vigilance.

R. E. Newton:

I am very pleased with some of the features of this discussion. I think it is fairly clear that we ship hydrodynamicists are not asleep in this matter. In case anybody has any doubts on the matter, they ought to know that there has been a presentation of a theory, much on the lines of this paper, employing servomechanics, by L. J. Rydill, who gave a paper in the Transactions of the RINA of 1959. It was a very thought-provoking paper and accounts for *transient* effects that the author has not referred to.

Let me digress a moment and agree with Captain Saunders. Quite frankly, sir, I find it difficult also to follow the jargon, and also, to be more frank, I find it very difficult to understand Rydill's paper.

It is a very important thing that the hydrodynamicists should take a lead from the aerodynamicists, and I honestly think that he does this every day. But when one turns to the study of ships he is dealing in a body in two media, not one, and this complicates the subject no end. I wonder, myself, what the automatic system is going to be in a ship, to steer it without any attention by the helmsman, the captain, or anybody else, when one takes account of pitch, surge, sway, and all this. Some attempt is being made to do this, as you know. In the first phase we are going to pose a submarine problem where one gets a break, and I feel sure that TMB, AEW, and probably Japan and others are already doing computer studies of this very problem.

Another impression that I thought I detected in the paper was that the ship was a very big thing and has plenty of room and weight to spare. Whether the impression was right or not I do not know, but I would like to say this—that the ship might be big, but there isn't a lot of weight and space to spare.

In other words, to sum it up, my opinion is that there is a lot to learn from this rather excellent paper by Segel but I think we have to be very careful about how we apply it to a ship. In the words of Lord Kelvin, "When you can measure what you are speaking about and give it specific numbers, then you do know something about it," and it is that stage which we have to reach in this course of study.

A. Suarez:

I have one point to make concerning the terminology and definition of handling qualities. Actually a ship does not go out to sea and turn around in circles and perform big fancy maneuvers. We have really two problems in the operation of vessels: one is maintaining course at sea, and the other the close-maneuvering problem associated generally with sheltered water (where we usually run into trouble).

The steering rules at sea are one thing, but the rules in sheltered water in the vicinity of walled bottoms and of other ships approaching is an entirely, drastically different situation. How any particular theory or rule can be incorporated into the operation of a vessel seems far remote at present, unless we develop a terrific amount of gear to put on a vessel (transducers all around to integrate the pressure on the vessel) which will tell the steering apparatus how to maneuver the vessel under any set situation. I don't see at present that we are going to be very successful along these lines. We are still going to have to rely on the judgment of the pilot, with his experience and background, running from one side of the bridge to the other, to bring the ship in from the ocean to a sheltered area.

Author's Closure:

Well, gentlemen, I expected controversy, and I must say that it certainly took place. I don't know how I can really do justice to a rebuttal here. It seems to me that it would take quite a lengthy period. I will try to do the best I can from the notes I took while the discussers were making their remarks.

I will start in order with a reply to Captain Saunders. I want to apologize for not using the proper language in my paper to do justice to some of these ideas and concepts that I wanted to present. I truly feel that this is regrettable, but I hope that in time we can overcome this language barrier.

Captain Saunders made mention of the role of the pilot and drew attention that the type of closed-loop system that I pictured in Figure 1 of my paper ignored the presence of other people and other factors in the loop. I agree with this comment; I believe it shows that if the responses of ships were not so slow; the physics of ships and water were not so complex, that ship-control procedures could not have evolved in the manner that they have today, in which intermediaries are introduced to effect the tightness of the loop. Now I realize that when I use the term *tightness* I again use a terminology which perhaps does not have meaning to many of you here. But the loop is not tight as it is when you are driving an automobile or perhaps flying an airplane, and therefore there is a fundamental difference between the fast-responding systems I have referred to and the ship-control problem. I think there is no question that pilots are necessary because very often the man at the helm is not in a position where he can see what is going on when he is trying to negotiate a channel. The pilot is his second pair of eyes running back and forth, as I think Mr. Suarez mentioned, to check where the ship is going, how close he is to the shore, and so forth. Perhaps in the future, if things eventually

go in the direction I have indicated they might, there will have to be a large amount of design consideration given to the helmsman, if he is to be the primary controller, and locate him where he can see and not need this extra pair of eyes.

Captain Saunders also asked some questions about Figure 3. I want you to notice that Figure 3 is plotted on the same coordinate and abscissa scale that is on Figure 1. Figure 3 shows the dynamic characteristics of what I choose to call the directional mode of an automobile. Actually an automobile turns out to be a fourth-order dynamic system because of the presence of a roll degree of freedom. Ignoring that extra degree of freedom and thinking only of the freedom to move laterally and to turn, you end up with a second-order system with a directional mode which is the mode that is primarily excited by the action of the steering wheel. This is merely a plot of the natural frequency versus the percent of critical damping of the second-order dynamic system, and you can think in terms of a damped system with one degree of freedom, but in this case we happen to be dealing with two degrees of freedom. These characteristics will describe the nature of the response: how rapidly it will build up; whether there is an overshoot; what the damping is like; and so on. I grant you, when a person drives an automobile he is completely unaware, in many instances, of the fact that there are appreciable dynamics between the time he turns the steering wheel and succeeds in turning the car. Actually, steering systems are not perfect. There are many lags; but if the front wheels of the car are controlled directly, i.e., the fixed control response of the car as assumed by Figure 3, you would find that this stable vehicle would have a response time of approximately 1 sec at 60 mph, where response time is considered in the manner that I defined it.

In answer to Mr. Lum, who made the excellent point that we do not know enough about the ship's transfer function, I confess that I am not really in a position here to comment as an expert as to whether we do or not. I see others around me who, with the facilities we are going to look at today, represent the capability of doing more about solving this problem. Getting additional information to solve the problem of representing strictly the hydrodynamic characteristics, complicates the picture to a very large degree in comparison with other vehicles. I think this is recognized. Many people have tried to examine the ability to control the ship by strictly linear mathematical models which are valid within limits, and certainly Norrbin's paper is an excellent textbook-type of summary of the situation as it exists today. I also felt that on examining many of the sections in Captain Saunders' excellent two volumes we are not in as bad a situation as perhaps Mr. Lum implies. I may have misinterpreted. Maybe he doesn't think we are in as bad a way as I have indicated. I agree with the rest of his remarks.

Mr. Williams has expressed some very nice things about my paper, and I would like to thank him for that. I don't believe that any comment is indicated.

Mr. Goodman, of course, has put me to the task and I'll try to do the best I can. He made reference to the display system that is omitted in Figure 1, and I agree with him wholeheartedly. The question of display ties in quite closely with the point raised by Captain

Saunders regarding the pilot.* These are the connections between ship response and the controller. The display represents a means whereby a man uses his senses to determine what the ship is doing in a given situation; for example, a submarine where there is not contact with the outside world and a man depends wholly on his instruments. I believe, as far as surface ships are concerned, when passing through channels there certainly is no replacement for the pilot's visual reference, and this would take precedence over any instruments that he may have on board. I agree with Captain Saunders that out at sea the gyrocompass is probably his primary reference. But to say a few more words about the display, many people, particularly the psychologists, concern themselves with the problem of improving vehicle handling qualities by making improvements on the display. This has been particularly true in the submarine field. I do not take issue or quarrel with them; in some cases there is much room for improvement.

The remarks in my paper were primarily directed toward what can be done to improve handling qualities and thereby improve overall ship maneuverability by working on the control system between the operator and the ship itself. To do justice to this argument would require a lengthy exposition. Briefly, these automatic control systems that I refer to as response augmentation systems, are, effectively, systems that are inserted between the helm and the rudder, or any other force-actuating device that may be installed on the ship. the purpose being to take over the response augmentation task that the helmsman tries to accomplish. I have argued that it may be possible to improve this system by allowing the helmsman to approach the simple controller task and be represented as a simple amplifier, and thus take a step in the right direction. To debate how far one can go in this direction gets into a wealth of engineering considerations. I should make it clear at this point that everything in my paper to all intents and purposes ignored the engineering problem. I am sure many of you have the engineering problem uppermost in your mind when you think about how you would implement some of the things that I have mentioned.

The definitive maneuver concept was mentioned as a means of assessing handling qualities, and I want to make it clear that I have no quarrel with this concept. I tried to make it clear in my paper that I merely wanted to introduce a rather drastic hypothesis as to what would constitute a definition or criterion of handling qualities, to underscore the importance of bringing the response characteristics of the human operator into the picture. Definitive maneuvers are a measure of ship behavior. The difference in the approach that I have indicated is that in my own opinion the most important variable in the handling qualities problem is the dynamics of the controlled element, bearing in mind the nature of the human operator; the dynamic characteristics of the controlled element form the fundamental variable in the whole picture. As far as the definitive maneuver goes, it involves a multitude of variables including the hydrodynamic characteristics of the ship and the characteristics of the control system. The maneuver is designed to bring out certain characteristics but these characteristics are related in a very complicated fashion to the dynamic response characteristics of the controlled element. I want, of course, to include the steady-state characteristics, the

steady-state relationships between the controlled displacement, or input, and steady-state response, in this case primarily turning response.

I agree that an attempt to interrogate operators has many pitfalls and would lead to failure. Our experience in the aircraft field has been that the only way we can get intelligent operator opinion is to make available to them an aircraft that can be flown and which has the capability of being altered rapidly from one dynamic response configuration to another. At the same time the operator is given tasks which are representative of what aircraft daily are required to do. For example, just fly straight and level, or attack a target. Fighter aircraft, we all agree, do have this one important objective.

I'll skip around here with respect to ships. Relative to making an assessment of the handling qualities of a ship, I tried to indicate that comments should be obtained from the pilot with reference to situations that are representative of what the ship controller has to deal with every day.

Mr. Goodman made reference to the point that you cannot get more out of the ship, in terms of maneuvering capability, than is in it, as for example, in terms of the force-producing characteristics of the hull and its rudder, and he is certainly right. This automatic control or response augmentation system I mentioned certainly cannot make the ship respond any faster or do anything more quickly than what control power is available to the ship by virtue of the size of the rudder and so on. I think there is a little misinterpretation; I was trying to indicate that the response augmentation system would merely try to take advantage of what is available in the ship in terms of control power. In this respect, it would do the best it can by moving these surfaces as rapidly as the steering motor will allow and by properly timing the whole operation where the precision of timing is increased by an order of magnitude over what the operator can do. He depends on his memory and his opinions to time his actions very carefully. The automatic control system depends on instruments which measure what the ship is doing and, by virtue of computers or other elements, causes the system to do the proper things automatically, thereby relieving the helmsman of the burden to speed up this time. Now, if research shows that, with the present engineering state of knowledge, handling qualities could be improved by an order of magnitude merely by introducing a response augmentation with the control elements that are presently installed in ships, then there would be reason for doing something along these lines. There would be reason for thinking in terms of propellers at the bow, perhaps at the stern. In more serious terms, we are concerned primarily with the maneuverability of the vessel in various situations and eliminating the need for tugs and so on which, as pointed out, can be a sizable and economical gain.

The question was raised as to where does the man fit into the automatic controls. I trust that my very inadequate remarks up to now have shed a little light on this. This automatic control is installed between man and the ship, and serves the function of allowing him to act as much as possible like a simple controller.

The question was asked, how does response augmentation improve stability? A response augmentation system involves a force-producing mechanism which produces forces proportional to perhaps the drift angle of the ship, the angle yaw of the ship, or some other motion variable. If I produce a force and moment on the ship by artificial means, proportional to some of these motion variables, I am basically changing the effective stability derivative of the ship. I am effectively changing damping in yaw. We all recognize that if we increase damping in yaw, we increase directional stability, and we are providing an effective improvement in the natural dynamic stability the ship possesses on its own without any active control element. The question asked further, why full-scale handling qualities tests? I don't believe I could do as much justice to answering this question as has been done in one of the sections in Captain Saunders' Volume 2. I recall reading a discussion of how the time scales differ in the model tests situation and the full scale. Finally, the last item I can comment on with regard to Mr. Goodman's remarks concerned increased rudder power and engineering considerations. In my ramblings, I said previously that engineering considerations have been overlooked. All I can say is it has been our experience in the aeronautical field that unless one is willing to overlook some of the problems one must face to achieve a particular objective the state of the art is not really advanced at a very rapid rate. We all know that increased performance has been the motivating impetus for forcing engineers to follow thousands of problems. and I am perhaps naive enough to think that the day will come, to think in terms of greatly increased performance being available on ships and submarines. I am sure it is coming, and these increases in performance are going to require considerable increases in controllability, stability, general handling qualities in order for the Navy to take advantage of these performance improvements. I want you all to understand that this last statement does not imply any kind of criticism as to whether we have good handling qualities at the moment in some of our latest subs or whether we do not. I think everyone has been doing an admirable job and the fact that these things are running around today is proof that the engineering problem has been solved. No one wants to sell anybody on automatic control. I merely want to lay open a vista here, the possibility of thinking along research lines, going out "into the blue."

Mr. Morse, from Sperry, made some remarks about my discussion of human behavior. By and large, I accept most of the refinements that he suggested. I want to make clear that this entire discussion of human dynamics, human response characteristics, has been extremely simplified for this presentation. As you all will recognize, there is a wealth of new ideas or concepts that I have tried to introduce. I, for one, could not do justice to all of this in a rigorous fashion in the time allotted.

Mr. Newton, as I recall, made some remark to the effect that some of these considerations on stability and control and damping have been [illegible] the mill. I would repeat again that I was primarily concerned with maneuverability as a [illegible]pical concept and the role of the human operator in this system problem, and have there[illegible] postulated some rather drastic assump-

tions as to what could possibly be used, for the time being, as measures of handling qualities for the objectives that I have indicated previously. Again in connection with the engineering problem raised by Mr. Newton, I think my remarks made with regard to Mr. Goodman's comments apply here as well.

EXPERIMENTS ON SHIP MANEUVERABILITY IN CANALS AS CARRIED OUT IN THE PARIS MODEL BASIN

by

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ABSTRACT

The paper describes the installation and the experimental techniques used in the Paris Model Basin for the investigation of ship maneuverability in canals. It reviews some of the results obtained from an important series of tests: the existence of a critical speed from the point of view of ship maneuverability, the influence of the dimensions of the ship and of the canal, the influence of the rate of change of rudder, and a comparison between a single-screw ship and a twin-screw ship.

I. INTRODUCTION

Ten years ago when twin-screw tankers of 30,000 tons deadweight were being built in France, we had indeed reason to worry about the way they would behave in the Suez Canal since ships of smaller dimensions had the reputation of steering very badly there.

Hence, the Paris Model Basin was asked to investigate this problem: a special test installation was set up* whereby it was possible to insure that the tankers in the design stage had adequate maneuvering characteristics.

The tests carried out on different ships with two propellers led to some interesting results,[2,3] in particular, they have shown the advantage of keeping the rudder well clear of the hull by the use of appropriate stern shapes (aperture between skeg and rudder) and they have also shown the advantage of increasing the rate of change of rudder.

Since 1954, the test installation has been enlarged and an important number of tests have been carried out on several ships, most of them fitted with a single propeller. The present paper describes the test installation presently used and gives a summary of the principal results obtained.**

II. DESCRIPTION OF EXPERIMENTAL ARRANGEMENTS

1. CANAL

Tests are carried out in a rectilinear basin whose dimensions are as follows:

Length:	155 m
Width:	8 m
Depth:	2 m (the depth of water is adjustable)

*The description of this test installation and of the test procedure used is the subject of Reference 1. References are listed on page 189.

**Maneuvering tests in a curvilinear channel have also been undertaken; these, however, will not be dealt with in this paper.

At one end, the basin is provided with a wave absorber.

Banks can be installed on the bottom of the basin over a length of 100 m; their distance apart as well as their slope are adjustable. They are composed of sheet-metal plates of 1-m length joined together; a small clearance exists between two successive plates and between the plates and the bottom of the basin, despite the precautions used when installing them. Hence, there exists between the inside and the outside of the canal a leakage which in fact increases its width; the maneuverability is thereby facilitated, but in a manner which we consider as very minor; it may be observed, moreover, that since the maneuvering characteristics of a model in a canal are probably not as good as those of the full-scale ship, the fact that the canal is not watertight tends to reduce the difference between model and ship. The channel is not closed at its ends; the full-width portions of the basin are used to accelerate and to stop the model.

2. MODEL

The model is self-propelled: however, in order to allow the propellers to work under conditions which are identical to full-scale conditions, a correction is applied by means of an aerial propeller placed in the centerline of the model.

The model is controlled from the carriage of the basin by means of a "fishing line" device.

3. PILOTING

The problem of piloting presents a certain difficulty when working with a free model. In order to bring the model back to the centerline of the canal, we actually have but one parameter at our disposal, i.e., the rudder angle, while the position of the model with respect to its ideal position is characterized by two parameters which are, for instance, the heading and the lateral deviation of the center of gravity. The adjustment of an automatic pilot is then practically impossible.

The piloting is carried out by eye by an operator placed on the carriage. The follow-up between the "helm" and the rudder is of the self-synchronous type; its accuracy varies between ± 2 deg approximately; the rate of change of rudder is constant, but may be adjusted before each test.

The piloting of the model presents numerous analogies to that of the full-scale ship; nevertheless, there exist some differences which may have an effect on the ease of handling, in particular:

1. Yawing motions of the model are, indeed, more difficult to estimate by eye than by means of a heading repeater. On the contrary, lateral deviations are easier to see on model, a centerline being painted on the bottom of the basin.

2. The speed of response of the pilot has a relative importance which is greater than for the full-scale ship since the "natural response time" of the model is shorter than that of the full-scale ship (in the ratio of the square roots of the lengths).

These differences show that the pilot of the model must have excellent reflexes and that he must be very well trained. This is also indispensable in order that we may be able to make valid comparisons between runs carried out under different conditions. All the tests considered here (about a thousand or so runs) were carried out by the same operator; the agreement of the results is particularly satisfactory and in almost all cases it may be said that any abnormal scatter is due to a hydrodynamic cause rather than to a piloting error.

4. MEASUREMENTS

The following measurements are made during each test run:

Speed V: this is the speed of the towing carriage,

Rudder angle α,

Heading K: this is furnished by a gyroscope,

Lateral deviation ϵ of the center of gravity of the model with respect to the centerline of the channel: a piece of mobile equipment located on the platform of the carriage, furnishes a narrow light beam parallel to the centerline of the basin; with the aid of a wheel which is manually operated, the light beam is focused continuously on a vertical mast placed in the center of gravity of the model.

The rudder angle, the heading, and the lateral deviation are simultaneously recorded as a function of time; Figure 2 gives an example of the recording. The torque on the rudder stock is also recorded as a function of time, but separately.

III. EXAMINATION OF SOME OF THE RESULTS

1. CRITERIA OF MANEUVERABILITY

It is not a question here of examining how we can positively measure the maneuvering characteristics of a ship in a canal, but only one of seeing how we can best utilize the measurements carried out in order to compare runs made under different conditions.

A. The Model Passes Through or It Does Not

To avoid damaging the model in the case where it would strike the banks, safety rails are installed all along the canal; four vertical rollers are placed on the model in front and in back. In this manner, it is easy to determine whether the model strikes the banks. A run may then be rated good if there has been no contact with the rails and bad in the opposite case. This first criterion has the disadvantage of being purely qualitative, but it generally cross-checks quite well with the other criteria and it is very useful, in particular, when we want to compare different ships with one another.

B. Maximum Values of the Rudder Angle, of the Heading, and of the Lateral Deviation of the Center of Gravity

These three values are not independent, but the relationships linking them with one another are not rigid; they depend a great deal on the reactions of the pilot who may, for instance, make it a strict rule either to make rudder movements of small amplitude or, on the contrary, to bring the model back to the centerline of the canal as quickly as possible, even though he has to execute a large rudder movement.

This criterion involves the risk of inflicting a penalty on a run in the course of which a piloting error has occurred since this will lead to abnormally large values, especially for the rudder angle.

C. Mean Values of the Rudder Angle, of the Heading, or of the Lateral Deviation of the Center of Gravity

The mean values are evidently those of the absolute values of the rudder angle, of the heading, or of the lateral deviation. They are not distorted to the same extent as the maximum values, in the case of a single piloting error.

D. Other Criteria May Be Contemplated

We may, for instance, seek to characterize the response of the ship to a given movement of the helm, but this response depends on four parameters (k, dk/dt, ϵ, and $d\epsilon/dt$ at the moment of the rudder movement) and it seems to us to be difficult to make this evident without carrying out special tests.

Finally, we believe that the criteria defined above are those which enable us to best determine the maneuvering characteristics in a canal for the tests carried out. They may be used simultaneously, but the ones which appear to us to be the most interesting are the mean rudder angle and the proportion of bad passages.

2. EXISTENCE OF A CRITICAL SPEED FROM THE POINT OF VIEW OF MANEUVERABILITY. EFFECT OF THE DIMENSIONS OF THE SHIP AND OF THE CANAL

A. Observations

One of the most important results of the series of tests carried out is to show that for a given ship and canal there may exist a critical speed from the point of view of ship maneuverability, i.e., a speed for which the difficulty of passing through the canal is the greatest. This is reflected in the increased percentage of bad runs, in an increase of the rudder angles (the maximum angle and mean angle) and, to a lesser degree, in an increase in the heading and in the lateral deviation.

The sharpness with which these phenomena manifest themselves depends on both the ship and the canal. Figure 3 gives an example for a tanker A where the critical speed does not exist, at least within the range of the speeds investigated; the ratio $S/\boxtimes$ of the section of the canal to the midship section area is large (9.49) and the passage is easy: the mean rudder angle $|\bar{\alpha}|$ exceeds 4 deg but a single time while the maximum rudder angle exceeds 7 deg only two times (absolute maximum: 13 deg).

Figure 4, on the contrary, gives, for the same tanker, an example ($S/\boxtimes = 3.63$) for which the critical speed is very pronounced: in the critical zone the model strikes against the banks rather frequently, the mean rudder angle $|\bar{\alpha}|$ is oftentimes between 10 and 15 deg, and the rudder very often is turned until it comes to the stop ($\alpha = 30$ deg); for speeds smaller or larger than the critical speed, the transit conditions improve appreciably.

In Figure 5 the zone of the critical speeds is plotted as a function of the test conditions for three different ships (A, B, and C)* which have comparable dimensions.

A is a tanker with a single propeller which was tested at different drafts in a canal whose channel bed was 51.30 m wide and which has a slope of the bank of one-third and a variable water depth.

B is a tanker with a single propeller which was tested in the same canal and in a narrower canal (width of channel bed 42.00 m).

C is a tanker with twin screws which was tested, at two drafts, in the first canal and in a wider canal (channel bed 76.40 m wide).

Despite the difficulty of determining precisely the zone of critical speeds in certain cases, the agreement between the various results is good. Of course, if the results have been plotted as functions of the $S/\boxtimes$, this does not mean that this is the only parameter to be taken into consideration; it would be necessary, in particular, to take into account the depth of the water H in the canal, which determines by means of $\sqrt{gH}$, the variations in the state of flow. It is, of course, true also that it is not necessary that the results of different ships, even similar ones, be identical.

B. TENTATIVE ANALYSIS

We may attempt to explain the existence of a critical speed, at least in the case of a ship with a single propeller, for which the rudder efficiency and the course stability of route depend very largely on the conditions of the propeller operation. When the speed is small, the effect of the restricted water does not make itself felt, the state of flow is steady, the disturbances are not very numerous, and they are all the easier to correct since the "lost times" (reflex times of the pilot, times of rudder change) are small with respect to the "natural response time" of the ship (which is the time, for instance, which it takes for the ship to travel its own length).

When the speed increases, there comes a moment when the flow is no longer steady; then there are increased risks of yawing motions resulting in a difficulty of passing through the canal, which finds expression in an increase of the rudder angle and in the number of bad passages.

*See Section 4 for the principal dimensions of these three ships.

If the speed increases even more while remaining in the zone of unsteady flow, the resistance becomes very great and the propeller works with an increased slip ratio which is favorable to the rudder efficiency; there occurs in that case a reduction of the rudder angle and in the number of bad passages.

Hence, there may exist a speed, close to the speed of the change of flow, for which the difficulties of passing through the canal are increased. Hence, for a given ship, the greater the effect of the restricted water, the lower the critical speed from the point of view of maneuverability. This is indeed what Figure 5 shows; several propulsion tests carried out for the tanker A show, moreover, that the critical speed occurs at a slightly lower speed than that for which there is a very sudden increase in the resistance.

In the case of a ship with two propellers, the change in the state of flow is very certain to produce disturbances; however, we have not carried our tests far enough to state positively that beyond the critical speed, the influence of the propeller makes itself felt sufficiently to reduce again the difficulties of passing through the canal.

3. INFLUENCE OF THE RATE OF CHANGE OF RUDDER

One of the results shown in the course of the tests carried out several years ago on twin-screw ships was concerned with the advantage of increasing the rate of change of rudder.

In the course of the series of tests with which the present paper is dealing, comparisons were made with variable rates of change (4 and 8 deg/sec for the full-scale ship in one case, 2.5 and 5 deg/sec in another case); these concerned single-screw ships.

The conclusion drawn previously has not been confirmed. This is easy to understand if we examine the recordings themselves (see Figure 2): the time during which the rudder is in motion is actually a small fraction of the total testing time and one would change practically nothing in the curves $\alpha(t)$, hence in the phenomenon as a whole, by adopting a very large rate of change.

Therein lies, probably, the explanation for the disagreement between the tests carried out on ships with a single propeller and those with two propellers.* In the case of a single-screw ship, the hull and the rudder form a rigid hydrodynamic unit and the time for setting up the circulation is of the order of magnitude of the time which the ship takes to travel its own length; it is always greater than the time for changing the rudder;** reducing this time would no longer hold any advantage within a certain limit.

In the case of a twin-screw ship, on the contrary, if the rudder is well clear from the hull, a circulation is set up about the rudder at the end of a period of time which is of the

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*Tests recently carried out on a tanker with two propellers have yielded a negative result; however, the number of passages with a variable rate of change was too small for any valid conclusion to be drawn.

**For L = 200 m and V = 5 m/sec, the time for setting up the circulation would be of the order of 40 sec while the time of change from −30 deg to +30 deg is only 15 sec for a rate of change of 4 deg/sec.

order of the time which the rudder takes to travel its own length. The advantage of increasing the rate of change of rudder is then a greater one.

Let us finally take note of the fact that, even if the model results are negative, it is probably advantageous to reduce the time of change of rudder for the full-scale ship since this is a lost time which is more important than the lost time due to the pilot, whereas respecting the model, these two lost times being of the same order,* the effect of the change of the speed of rudder movement runs the risk of being obscured.** The torque on the rudder stock always remains smaller than the torque in unrestricted water (due to the reduction of the normal speed); the increase in the rate of change of rudder, in the canal only, does not, moreover, lead to an increase of the power of the steering gear.

4. COMPARISON BETWEEN A SOLUTION WITH ONE PROPELLER AND A SOLUTION WITH TWO PROPELLERS

A single ship with two propellers was tested in the course of the series of tests which are being considered here: this is the tanker C. Its forms are derived from those of the tankers of 30,000 tons for which the first test facility had been constructed; hence, it was especially designed for the maneuverability in a canal. It has a semibalanced rudder, whose upper portion pivots about the after edge of a skeg with a horizontal base; this arrangement is favorable to the rapid setting up of the circulation about the rudder since it is well clear of the hull, and favorable for course keeping because of the extended skeg. In addition, the rudder area is indeed larger than usual† and its rate of change likewise (8 deg/sec in a canal).

The maneuverability of the tanker C may be compared to that of the tankers A and B with a single propeller and a balanced rudder which were tested in the same model canal under similar conditions of draft and of depth of water in the canal.

The table indicates the test conditions under which the three vessels may be compared. The absolute values given for A and B are those of the full-scale ships, whereas the values indicated for C correspond to a ship geometrically similar to the full-scale ship such that the canal is the same for the three ships†† (width of channel bed: 51.60; slope of the bank 1/3).

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*For a 1/30-scale model, the change of rudder from −30 deg to +30 deg is effected in less than 3 sec if the speed of change of the ship rudder is 4 deg/sec.

**It is possible [illegible] such a case that a good model pilot does not feel the need for a rapid change of rudder, and that the opposite applies for a second-rate pilot.

†Under the test condition 2_C (see table), the ratio $\frac{\text{projected area of the rudder}}{\text{longitudinal area of the hull}} = 0.029$.

††In the same model canal, we have actually tested ship models on different scales (1/32.5 for A and B, 1/29.8 for C).

The condition 1_C may be compared to the condition 1_A; moreover, it is *a priori* less difficult (larger rudder area, greater rate of change of rudder,* larger $S/\boxtimes$). The ships A and C give rise to similar phenomena for the conditions 1_A and 1_C, respectively: up to 8 knots, the passage is relatively easy, but from 8 to 9 knots (maximum speed of the tests), it becomes very difficult: α_{max} = 30 deg; and the proportion of bad passages is high: 1/3 to 1/2 for 1_A; 1/1 for 1_C.

Fixed Conditions	Variable Conditions: Number of Test	Variable Conditions: Draft (Ship) m	Variable Conditions: Depth of Water (Canal) m	Variable Conditions: $\frac{S}{\boxtimes}$
Ship A				
Length between perpendiculars = 250 m	1_A	7.9	13.75	4.6
Beam = 34.1 m				
Rudder area = 47.0 m² / 50.5 m²	2_A	10.2	12.9	3.3
Rate of change of rudder = 4 and 8 deg/sec	3_A	10.7	12.9	3.2
Ship B				
Length between perpendiculars = 229 m				
Beam = 32.2 m	1_B	10.2	12.9	3.5
Rudder area = 41 m² / 48.0 m²				
Rate of change of rudder = 5 deg/sec				
Ship C				
Length between perpendiculars = 248 m				
Beam = 33.0 m	1_C	8.3	14.7	5.1
Rudder area = 80.5 m²				
Rate of change of rudder = 8 deg/sec	2_C	12.0	14.7	3.6

The condition 2_C may be compared to conditions 2_A, 3_A, and 1_B, but here again, and for the same reasons, it is *a priori* less difficult. For 2_C, the tests were carried out only between 5.2 and 6.8 knots: α_{max} = 30 deg, the proportion of bad passages is 2/3.

For 2_A, between 5.2 and 6.5 knots (critical zone for the maneuverability), α_{max} = 30 deg, $|\bar{\alpha}|_{max}$ = 12.5 deg; the proportion of bad passages is 1/10.

For 3_A, between 4.8 and 6.2 knots (critical zone), α_{max} = 30 deg, $|\bar{\alpha}|_{max}$ = 12.5 deg; the proportion of bad passages is 1/5.

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*4 deg/sec for the condition 1_A.

For 1_B, between 6.1 and 7.4 knots (critical zone), α_{max} = 25 deg, $|\overline{\alpha}|_{max}$ = 10 deg; the proportion of bad passages is 1/20.

The comparison between 1_C and 1_A, on the one hand, and between 2_C and 2_A, 3_A, 1_B, on the other, show that for difficult transit conditions, a good solution with two propellers remains inferior to a classical solution with a single propeller. In ballast-trim, however, the difference between the two solutions would generally be smaller than under load* since the single propeller, of large diameter, and the rudder then work under bad conditions (chances of emersion).

IV. CONCLUSIONS

One may ask if by giving an important role to a human pilot, one does not introduce a subjective element likely to distort the phenomena which one wishes to measure. However, the results obtained show that the method used enables us to form a correct idea concerning the qualities of a ship and even to establish objective laws which are confirmed with a reasonable approximation by all the ships tested.

A good example of this statement, it seems to us, is the obvious fact that there exists a critical speed from the point of view of maneuverability in a canal; for a speed somewhat below that of the "wall" of resistance, it is found that it is difficult to maneuver, whereas for lower or higher speeds, this difficulty disappears. Results obtained under these very different conditions agree pretty well with one another.

If, contrary to our expectations, the effect of the rate of change of rudder turned out to be practically zero, it is probably due to the fact that the tests dealt with single-screw ships for which the lift at the moment of the rudder change is not produced in quite the same manner as on a twin-screw ship. New tests will have to be undertaken to clear up this problem.

We have shown on the basis of an example what difference exists, from the point of view of maneuverability, between a ship with two propellers considered to be excellent and a ship with a single propeller of classic design. For difficult transit conditions, the ship with a single propeller proves to be the best in any case; this does not imply, however, that adequate characteristics are unobtainable for a ship with two propellers.

In this paper we have not been concerned with a comparison between the model and the full-scale ship. There is a lack of sufficient data in this connection; moreover, it is difficult to make a complete comparison because, apart from the hydrodynamic scale effects, there may exist scale effects which are due to the nonsimilarity of the lost times (rapidity of estimating the yawing motions, reflex times of the pilot, and possibly, times for transmitting the orders). We can state positively, however, that the ships which have been found to be satisfactory in the model basin have always proved to be very maneuverable in actual canals.

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*Up to 8 knots, 1_C and 1_A yield comparable results.

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3. Brard, R., "Maneuvering of Ships in Deep Water, in Shallow Water, and in Canals," Society of Naval Architects and Marine Engineers (1951).

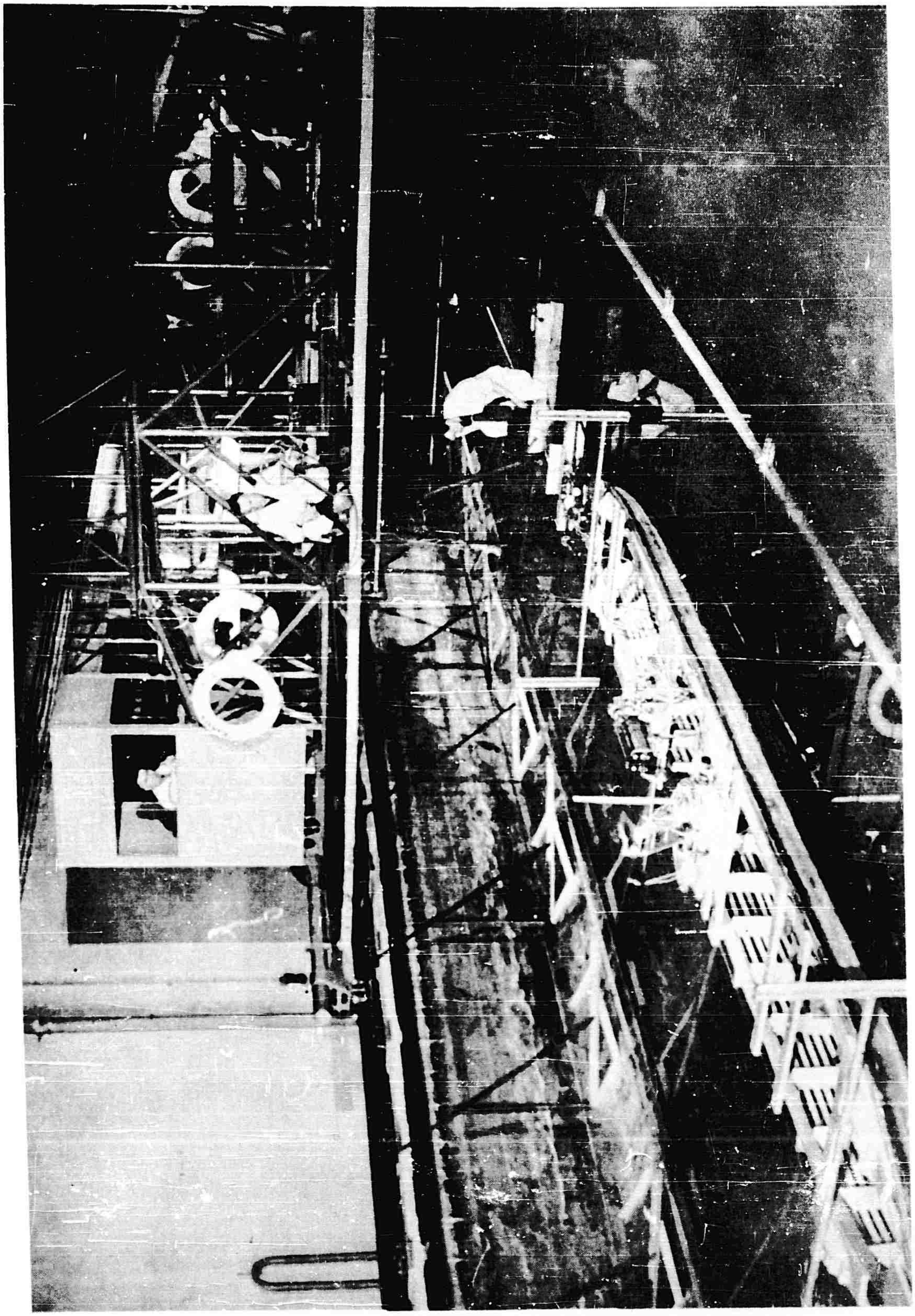


Figure 1

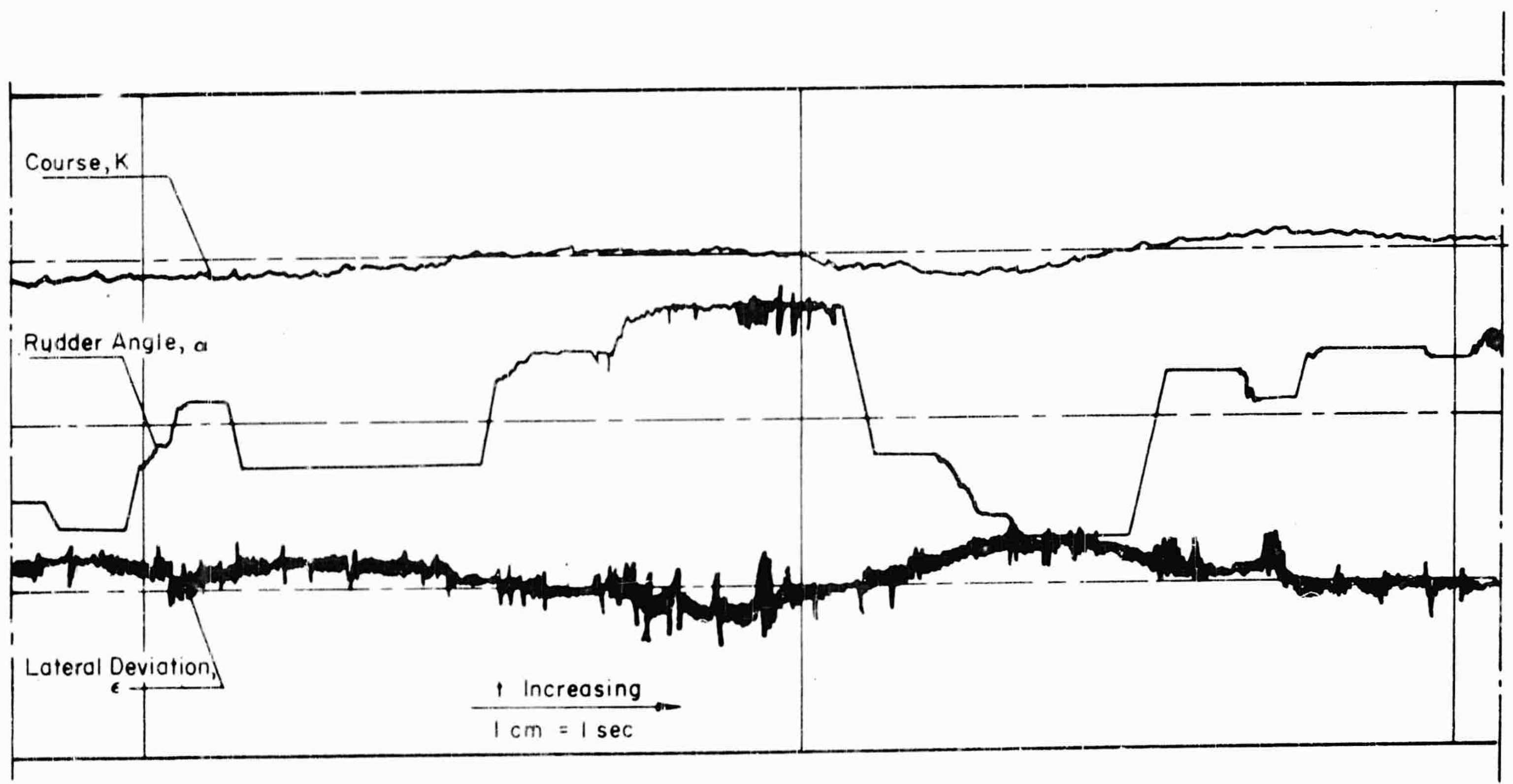


Figure 2 – Example of a Recording

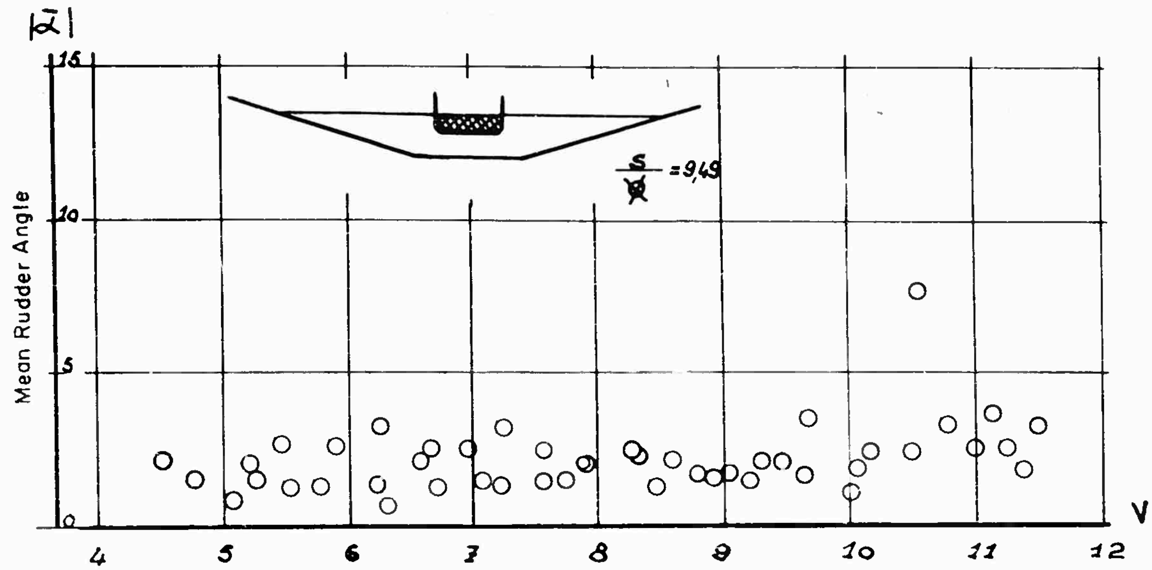


Figure 3

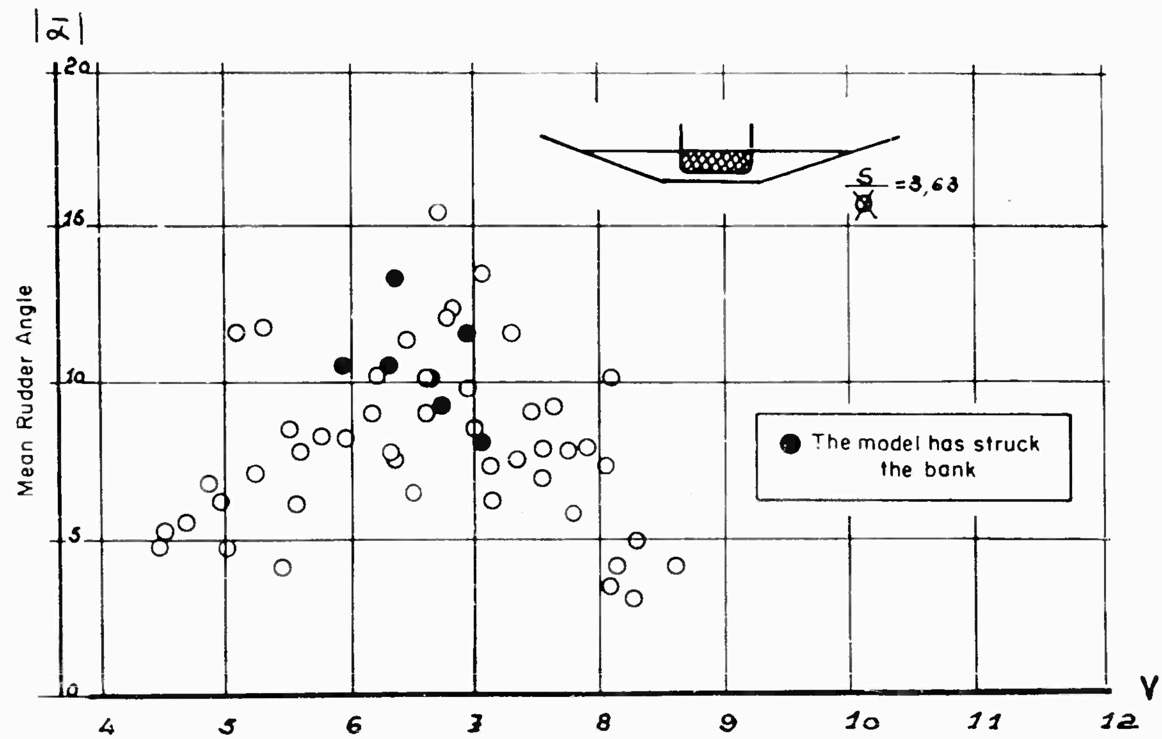
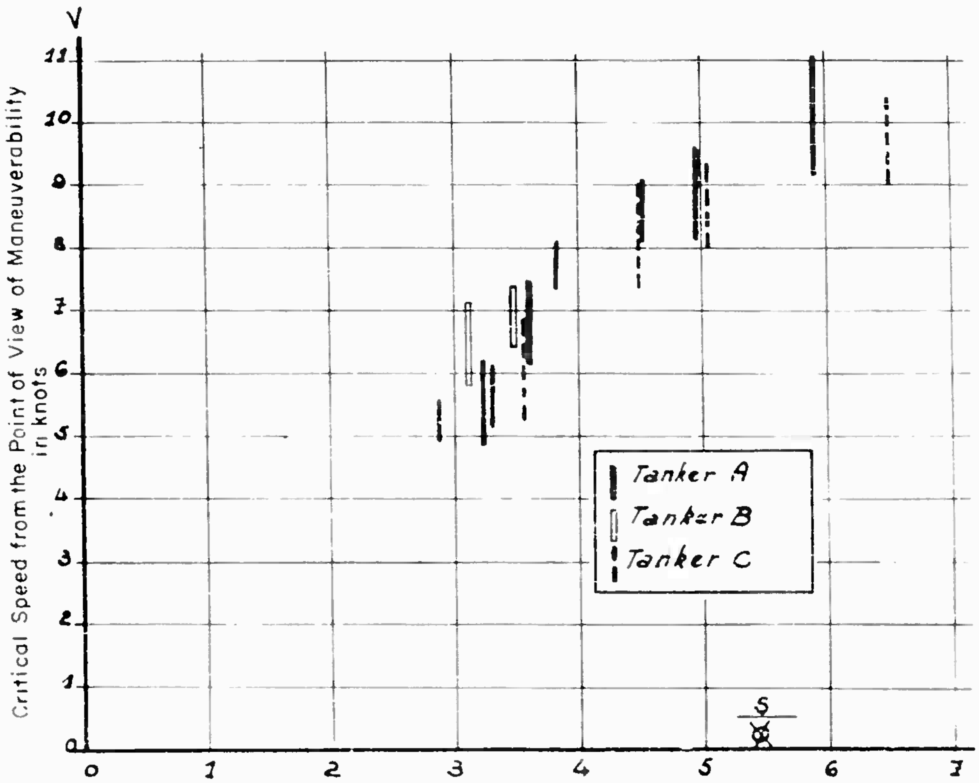


Figure 4



V
Critical Speed from the Point of View of Maneuverability in knots
0
1
2
3
4
5
6
7
8
9
10
11
Tanker A
Tanker B
Tanker C
S

Figure 5

DISCUSSIONS

H. E. Saunders:

I wish simply to point out one feature which I believe is responsible for the critical nature of the maneuvering of these models at the speed which you call the "critical speed." You do not give the water depth. I can't calculate the speed of the solitary wave but I am sure that your trouble comes when the ship is either rising up on the back of a solitary wave or getting over on the top of it. Your difficulty comes about by running at the speed of the solitary wave. The water depth should be somewhat in the order of a foot and a half.

Mr. Bindel's reply:

The critical speed for maneuverability coincides approximately with the change between steady and unsteady flow; it is below the speed of the solitary wave (which we never attained). As pointed out in the paper, the speed of change of flow depends not only on $S/$[illegible] but also on the water depth H by means of $\sqrt{gH}$.

H. E. Saunders:

I still believe there is some connection between the two.

V. E. Williams:

I would like to ask one question. In work that we did, sponsored by TMB, we found that the performance of a ship is greatly affected by the rate of the motion of the control surface and as a function of the stability of the boat. I would like to ask the author what is the relative stability of the boat.

Mr. Bindel's reply:

We did not measure the stability on course of the ships tested.

V. E. Williams:

The boat that is relatively stable will be less affected by the rate of motion of the control surface, whereas a boat that is very unstable will be greatly affected by the greater motion of the control surface.

S. T. Mathews:

I would like to thank Mr. Bindel for the excellent description in his paper of his shallow-water maneuvering facilities. I would like to ask Mr. Bindel if he considers any useful purpose could be brought about by breaking down the overall problem into a number of components. For instance, to use the model and canal for finding out the proximity to the bank required for grounding and then to measure the stability and control derivatives of the model in the horizontal plane under shallow-water conditions. Then I think the important next step is the standard analog computer, and it wouldn't be difficult to have a simulator rigged up for three degrees of freedom so that you could have a 1:1 time correspondence on the system. Then the problem narrows down to steering the vessel within known restrictive tracks. Furthermore, you could have perhaps a more realistic space system which was referred to this morning.

Mr. Bindel's reply:

I think it would be very interesting to divide the problem into components but the number of these is great and, therefore, is also great the number of tests to deal with the whole of the question. A significant advantage of our method is to enable us to find very quickly if a given ship is good or not. But, for systematic series of tests it would be useful to determine the effect of the various parameters, and, for that case, to keep in mind the feasibility of a simulator.

C. G. Moody:

Mr. Bindel has given us some interesting statistics on the handling qualities of ships in canals. His note on the existence of a critical speed is intriguing. No doubt it is asking a great deal, but in continuing with this work it would be very helpful if he could supplement these statistics with some pertinent observations on the accompanying physical phenomena.

The author has followed the best scientific tradition in showing all of the data points, including one lonesome point, in Figure 3. There are transient effects in canal tests that could easily make one data point different from the others. In this instance, though, the lonesome point may have some significance; for it is not only the highest point in the figure, but the lowest points in the same region are higher than the lowest points in any other part of the figure.

The ship speeds represented in these tests were well below the solitary wave speed for the approximately 13-meter water depth of the main channel. However, the wave pattern of the ship is affected along the sides of the canal by the shallow water over the sloping banks.

It may be of interest to note in this connection that a solitary wave can be created by the sudden entrance of a vessel into the confined water of a canal, or by the initial accel-

eration of a vessel that is set in motion within the canal. In some canal tests at the David Taylor Model Basin it was found that the starting movement of the model under the carriage created a wave crest about $\frac{1}{4}$ inch (or 6 mm) high which ran out ahead of the model down the channel, and also a wave hollow which simultaneously ran in the opposite direction astern of the model. A similar occurrence was reported by Rear Admiral E. A. Wright* (then Lieutenant Commander Wright) but in that instance the wave, which was observed in the Experimental Model Basin at the Washington Navy Yard, was only a few thousandths of an inch high.

The author attributes the fact that the handling qualities of the ship are improved when the speed is increased above the critical speed, as shown in Figure 4, to the greater propulsive thrust, which is favorable to the rudder action. A practical solution to the problem of handling such ships in canals consequently is to put a tugboat behind the ship to hold back on the stern and in this way to increase the propeller thrust of the ship at a given speed. The hawser should be fairly long to permit the tug and the ship to steer independently of each other, and generally should be attached to the tug at a point about 0.2 of the length from the bow. The point of attachment should be a little forward of the center of lateral resistance of the tug on a straight course in yaw; that is to say, it should be a little forward of the point that Mr. R. E. Newton has aptly referred to in his paper as the "neutral point." The paddle tugs of the British Admiralty are well suited to this purpose because they can back both paddles and still steer with the rudder in the normal fashion while going ahead. The idea of using a tug astern of a ship for this purpose has been previously presented.**

J. L. Goldman:

Mr. Bindel has mentioned the phenomenon of the twin-screw vessel benefitting from a very high rate of rudder change, that is, in the order of 8 degrees per second, whereas the single-screw ship, as I understand it, receives no benefit from this phenomenon. I wonder if the explanation that Mr. Bindel gave us is the correct one or if there is possibly a simpler one that explains it. Is it just that in the twin-screw ship, as the rudder swings over more rapidly, it comes more quickly into the screw race coming off from one of the propellers, and then the rudder becomes more effective. We have found so often that until the rudder gets into a propeller race on a twin-screw vessel that to fix a lower speed is not very effective at all. Perhaps he is accomplishing this. I would like to ask this as a question.

* Wright, E. A., Comments on "The Effect of Size of Towing Tank on Model Resistance," Transactions, Society of Naval Architects and Marine Engineers, Vol. 50, p. 187 (1942).

** Moody, C. G., "The Handling of Live Super Ships through Gaillard Cut of the Panama Canal," David Taylor Model Basin Report 1277 (Oct 1958).

Mr. Bindel's reply:

I think that the remark of Mr. Goldman may contribute for a part to the understanding of the matter. The explanation I gave concerns a twin-screw ship with a rudder well clear of the hull and well clear of the propellers; if you turn the rudder, the time to establish the circulation is of the order of the time for the *rudder* to travel its own length. On the contrary, for a single-screw ship the time to establish the circulation is of the order of time for the *ship* to travel its own length. Setting up of circulation on a twin-screw ship and on a single-screw ship is therefore different.

DATA FOR ESTIMATING BANK SUCTION EFFECTS IN RESTRICTED WATER AND ON MERCHANT SHIP HULLS

by

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ABSTRACT

In 1946-47 the David Taylor Model Basin conducted an extensive study on the performance of model ships in restricted channels primarily to obtain information which would be of assistance to the Panama Canal Company in the design of a modernized canal across the Isthmus of Panama. The results of this study were presented in TMB Report 601 by R.S. Garthune, B. Rosenberg, D. Cafiero, and C.R. Olson. Pertinent portions of these results were re-analyzed recently by the author in an endeavor to derive broadly applicable data for estimating the effects of bank suction in restricted water on full-lined merchant ship hulls. The basic data, although limited in scope, indicated consistent trends which made it possible to derive a simple nondimensionalized design chart for estimating the magnitude of side force and turning moment on ships having the general proportions of modern tankers, as functions of channel depth to draft ratio, channel width to beam ratio, and distance from the centerline of the channel to beam ratio. It is believed that this chart will assist the designer of the ship's rudder in making proper allowance for the additional side force and turning moment experienced by the ship when transiting restricted waterways.

INTRODUCTION

In the course of analyzing the causes for the grounding of several merchant vessels in the Panama Canal, the author had occasion to investigate the effect of "bank suction" in narrow canals on the controllability of transiting vessels. This effect is well known and easily explained in qualitative terms; however, quantitatively, the effects are not easily computed and, in general, must be obtained experimentally. Model tests lend themselves for this purpose and have often been resorted to.

An extensive investigation on model scale was carried out in the David Taylor Model Basin in 1946-48 at the request of the Panama Canal Company. The primary purpose of this investigation was to obtain data for selecting the best channel cross section and the characteristics of channel bends for a modernized Panama Canal but, as a byproduct, numerical data on bank suction effects were also obtained. The results of this investigation were published in TMB Report 601 by R.S. Garthune, B. Rosenberg, D. Cafiero, and C.R. Olson.

In view of the narrowly prescribed objective of the project, no attempt was made in Report 601 to draw broad conclusions from the data. Hence, when the writer in the aforementioned study attempted to use the data for estimating bank suction effects for ships of various sizes, a complete re-analysis of the pertinent data became necessary. This analysis and the results obtained are presented in the following.

GENERAL CONSIDERATIONS

When a ship or similar body proceeds at steady speed through a fluid at rest, fluid is continuously displaced at the bow and transported around the ship to fill the void behind the stern. This flow produces pressures on the hull which in unrestricted water are balanced, port and starboard, so that no forces tending to transport the ship laterally arise. The same will be true in a channel restricted in width as long as the ship advances along the centerline of the channel. However, it is no longer true when the ship advances parallel to but to one side of the channel centerline. In this case, the pressures on port and starboard are different in magnitude and distribution so that lateral forces and yawing moments arise. In general, the directions of these forces and moments are such that the vessel will be transported toward the nearer bank, and the bow tends to sheer away from the nearer bank. An exception may arise when the vessel has a large initial yaw. In this case, bank suction may increase the yaw angle through Venturi effect which may not be correctable by the rudder and may lead to the bow striking the bank.

MODEL TEST RESULTS

In the investigation referred to, several types of tests were run and several types of models were used. For information on these, the reader is referred to the original report. Amongst them were tests on a model of a large tanker performed in the TMB circulating-water channel in which lateral force and turning moment were measured. These are the only tests of interest here and will be discussed in the following.

The circulating-water channel of the Taylor Model Basin has a test section 22 feet wide, 60 feet long, and 9 feet deep; the water speed can be varied up to a maximum of 10 knots. Water depth can also be varied by running the channel partly full, and channel width can be varied by inserting removable partitions into the test section.

The tests of interest here were conducted on a model representing a twin-screw tanker having a length of 720.6 feet, 100-foot beam, 32.13-foot draft, and fitted with a single rudder. In these tests, the model was held stationary at zero yaw angle (parallel to the channel walls) with the rudder on the centerline and propellers running at self-propulsion rpm in streaming water varying in speed from 4.5 to 10 knots (full scale). The channel width and depth and the position of the model with reference to the channel centerline were varied. The forces and moments to hold the model in equilibrium were measured by suitable instrumentation. The data obtained, taken from Table 4.2 of TMB Report 601, are reproduced in Table 1. All dimensions in this table are given in terms of full scale.

ANALYSIS AND PRESENTATION OF THE DATA

The data measured in the tests were treated as follows. From the measured lateral force and water speed a nondimensional coefficient was computed; this coefficient is defined by

$$C_F = \frac{F}{\rho/2\,(L\,d)\,v^2}$$

where F is the lateral force in pounds,

ρ is the fluid density in slugs per cubic foot,

L is the ship length in feet,

d is the ship draft in feet, and

v is the ship speed relative to undisturbed water in feet per second.

Next, the distance Y from the center of the channel was expressed as a fraction of the ship beam B. Finally, the product of the lateral force F and of the ship length L was computed. The results of these computations are shown in the last three columns of Table 1.

The data given in Table 1 were analyzed graphically as follows:

a. The variation of C_F with distance from the channel centerline for constant channel depth was determined. This resulted in the graph, Figure 1, in which C_F is plotted against Y/B for a channel depth H of 45 feet, or for $H/d = 1.40$.

b. The variation of C_F with channel depth for constant channel width was investigated. This resulted in the graph, Figure 2, in which C_F is plotted against Y/B for a constant channel width of 500 feet ($W/B = 5.0$) and for three depths: 45 feet, 60 feet, and 80 feet or $H/d = 1.40$, 1.866, and 2.49.

c. The location of the center of lateral pressure on the hull was found by plotting the measured yawing moment M against the computed product FL. This resulted in the graph, Figure 3.

In each figure, faired curves were drawn through the experimental spots. It will be seen that the spots are considerably dispersed but nevertheless show consistent trends. This consistency enabled construction of the design charts, Figures 4 and 5. In Figure 5, C_F is given as a function of Y/B for constant values of channel width to beam ratios. Figure 4 gives the location of the center of lateral pressure $\bar{x}$ determined from the relation

$$\bar{x} = \frac{M}{FL}$$

and of the depth correction factor (α) applicable to C_F determined from the relation

$$\alpha = \frac{C_F \text{ for given } H/d}{C_F \text{ for } H/d = 1.40}$$

ILLUSTRATIVE EXAMPLE

The application of the design charts becomes clear from the following numerical example. Find the bank suction force and the yawing moment in a restricted rectangular

channel when the ship is advancing parallel to the bank but to one side of the centerline of the channel, assuming the following data to be given:

Channel width	$W = 400$ feet
Channel depth	$H = 45$ feet
Distance off channel C.L.	$Y = 120$ feet
Ship length	$L = 700$ feet
Ship beam	$B = 90$ feet
Ship draft	$d = 28$ feet
Speed	$v = 10$ feet per second
Salt water density	$\rho = 1.99$ slugs per cubic foot

From the given data, we get

$$Y/B = \frac{120}{90} = 1.33$$

$$W/B = \frac{400}{90} = 4.44$$

$$H/d = \frac{45}{28} = 1.61$$

Entering Figure 5, we find

$$C_F = 0.0390$$

and entering Figure 4 with $H/d = 1.61$, we get

$$\alpha = 0.915$$

$$\bar{x} = -0.226$$

Hence, the lateral force F is

$$F = 0.0390 \frac{1.99}{2} (700)(28)(100)(0.915) = 69{,}600 \text{ pounds}$$

and the yawing moment M is

$$M = (-0.226)(700)(69{,}600) = -11{,}000{,}000 \text{ pounds-feet.}$$

The minus sign indicates that the moment is counterclockwise when the force is considered positive pointing to starboard, and inversely.

TABLE 1

Lateral Force and Yawing Moment at Zero Yaw and Zero Rudder Angles for a Twin-Screw Tanker

$L = 720.6 \times B = 100 \text{ ft} \times d = 32.13 \text{ ft.}$

Run No.	Channel		Side Slope	Speed knots*	Y** feet	Yawing Moment† feet-tons	Total Lateral Force tons	Y/B	C_F	FL feet-tons
	Width feet	Depth feet								
1	268	45	Vertical	4.56	23.2	– 562	4.9	0.232	0.00826	3,530
2	268				46.6	– 3,495	25.7	0.466	0.0433	18,520
3	500				23.3	– 413	1.3	0.233	0.00219	936
4					45.8	– 413	3.0	0.458	0.00506	2,160
5					142.8	– 3,902	18.0	1.428	0.0303	13,960
6					142.8	– 3,845	16.1	1.428	0.0271	11,600
7					142.8	– 3,264	16.3	1.428	0.0275	11,740
8					142.8	– 3,208	19.9	1.428	0.0335	14,340
9					166.2	– 4,783	23.4	1.662	0.0395	16,850
10					189.4	– 4,990	36.0	1.894	0.0607	25,920
11				7.5	23.3	– 188	10.3	0.233	0.00642	7,420
12					45.8	– 2,138	10.7	0.458	0.00667	7,700
13					92.4	– 5,622	26.2	0.924	0.0163	18,860
14					92.4	– 5,121	26.8	0.924	0.0167	19,300
15					92.4	– 5,220	25.7	0.924	0.0160	18,510
16					119.5	– 8,415	41.4	1.195	0.0258	29,820
17					166.3	–13,970	79.2	1.663	0.0494	57,000
18				10.0	23.3	– 1,261	7.1	0.233	0.00249	5,120
19				10.0	45.8	– 4,374	22.9	0.458	0.00802	16,500
20				10.0	92.4	–11,052	45.7	0.924	0.0160	32,900
21				10.0	119.5	–17,482	63.0	1.195	0.0221	45,400
22	500	60		4.56	23.3	– 195	0.9	0.233	0.00152	648
23					45.8	– 435	3.6	0.458	0.00606	2,590
24					92.4	– 1,073	7.9	0.924	0.0133	5,690
25					119.4	– 1,500	12.9	1.194	0.0217	9,920
26					142.8	– 2,054	16.3	1.428	0.0275	11,740
27					166.3	– 2,700	22.3	1.663	0.0376	16,060
28					189.4	– 3,510	31.5	1.894	0.0531	22,680
29				7.5	23.3	– 128	7.1	0.233	0.00443	5,120
30					45.8	– 1,185	8.8	0.458	0.00548	6,340
31					92.4	– 2,895	21.4	0.924	0.0133	15,410
32					92.4	– 2,872	20.0	0.924	0.0125	14,410
33					119.4	– 3,248	31.1	1.194	0.0194	22,400
34					142.8	– 5,310	38.8	1.428	0.0242	27,930
35					166.3	– 7,159	55.6	1.663	0.0347	40,100
36					189.4	– 7,645	80.2	1.894	0.0500	57,800
37				10.0	23.3	– 997	10.1	0.233	0.00354	7,280
38					45.8	– 2,511	9.9	0.458	0.00347	7,130
39					92.4	– 5,510	37.3	0.924	0.0131	26,870
40					119.4	– 7,070	50.8	1.194	0.0178	36,600

TABLE 1 (Continued)

Run No.	Channel Width feet	Channel Depth feet	Side Slope	Speed knots*	Y** feet	Yawing Moment † feet-tons	Total Lateral Force tons	Y/B	C_F	FL feet-tons
41	500	60	Vertical	10.0	142.8	−10,370	75.8	1.428	0.0266	54,600
42		60		10.0	166.2	−13,850	105.8	1.662	0.0371	76,200
43		60		4.56	23.3	0	0	0.233	0	0
44					45.8	− 75	2.1	0.458	0.00354	1,513
45					92.4	− 360	8.4	0.924	0.0142	6,050
46					119.4	− 998	10.1	1.194	0.0170	7,275
47					142.8	− 1,447	13.5	1.428	0.0228	9,720
48					166.2	− 1,687	19.3	1.662	0.0325	13,900
49					189.4	− 2,383	24.2	1.894	0.0408	17,420
50				7.5	23.3	− 127	2.2	0.233	0.00137	1,586
51					23.3	0	1.3	0.233	0.00081	937
52					23.3	− 375	4.3	0.233	0.00268	3,100
53					45.8	− 622	7.1	0.458	0.00443	5,115
54					92.4	− 1,500	15.9	0.924	0.00991	11,450
55					142.8	− 3,000	30.0	1.428	0.0187	21,620
56					166.3	− 4,071	35.0	1.663	0.0218	25,200
57				10.0	45.8	− 825	15.0	0.458	0.00526	10,800
58					92.4	− 2,378	30.2	0.924	0.0106	21,750
59					119.4	− 2,685	46.5	1.194	0.0163	33,500
60					142.8	− 4,635	58.6	1.428	0.0205	42,200
61					166.3	− 7,875	73.0	1.663	0.0256	52,600
62	770	45		4.56	97.3	− 900	4.3	0.973	0.00724	3,098
63					143.5	− 825	4.9	1.435	0.00826	3,530
64					213.5	− 1,650	10.5	2.135	0.0177	7,560
65				7.5	46.6	− 1,000	6.5	0.466	0.00405	4,680
66					167.0	− 4,700	16.5	1.670	0.0103	11,880
67					213.6	− 7,050	27.0	2.136	0.0168	19,440
68					260.0	−10,000	39.5	2.600	0.0246	28,450
69				10.0	143.5	− 5,739	41.6	1.435	0.0146	29,970
70				10.0	213.6	−13,870	55.8	2.136	0.0195	40,200

* Ship speed relative to undisturbed water.

** Y is the distance between centerline of channel and ship's center of gravity.

† The minus sign indicates a counterclockwise moment when F is pointing to starboard, and inversely. Negative yawing moment indicates a tendency to sheer away from the near bank.

CONCLUDING REMARKS

In the investigation from which the data in Table 1 were taken, the lateral forces and yawing moments were measured with and without propellers running; the difference between the two sets of data was found to be small. From this it may be concluded that bank suction effects are not greatly affected by propeller action and that the data presented may be used for estimating bank suction effects for single-screw as well as for twin-screw merchant ships of relatively full form.

One restriction on the use of the given data for estimating bank suction effects is the fact that the data apply only for the condition when the ship advances parallel to but to one side of the centerline of the channel. As indicated previously, bank suction produces both a lateral force and a turning moment. Hence, for the vessel to be in directionally stable equilibrium, an equal and opposite force and moment must be applied. In general, the rudder alone is not able to supply this force and moment and the vessel must be allowed to take on a small yaw angle to supplement the rudder effort; in other words, the vessel must be allowed to advance crab fashion. Therefore, the conditions at which the tests were run are not equilibrium conditions. In view of this, it might seem that the data given are of little practical use; however, this view would be unduly pessimistic. The condition of the vessel advancing to one side and parallel to the channel centerline is an unfavorable one. Hence, if the rudder has sufficient power to counteract bank suction for this condition, in general, it will also be able to control the ship for other conditions. An exception to this may occur when the ship develops a large initial sheer and the moment developed by bank suction tends to increase rather than break the sheer. In this case, uncontrollable motion may result that may end in grounding.

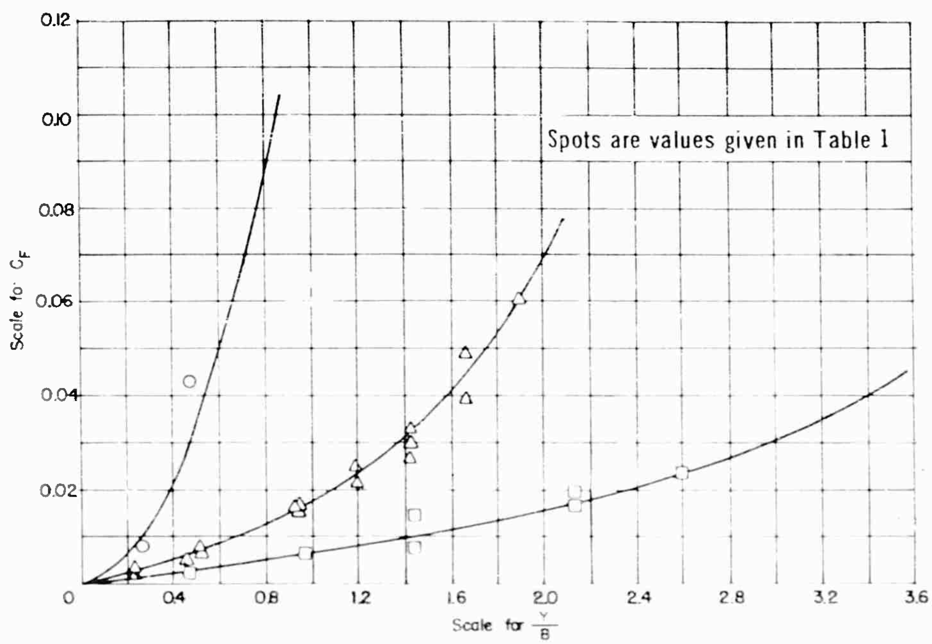

Figure 1 – Variation of C_F with W/B for Constant $H/d = 1.40$

$$C_F = \frac{F}{\rho/2\,(Ld)\,v^2}$$

F = Lateral force in pounds

ρ = Fluid density

L = Ship length in feet

d = Ship draft in feet

v = Ship speed in feet per second

Y = Distance from channel centerline

B = Ship beam in feet

W = Channel width in feet

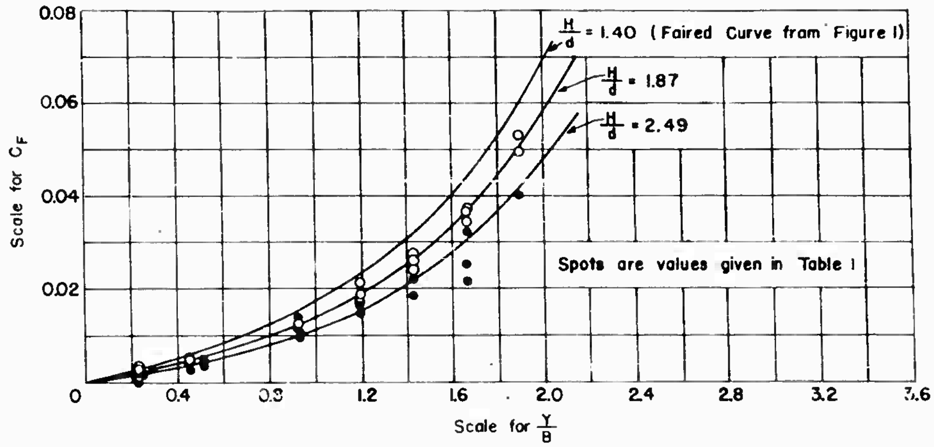

Figure 2 – Variation of C_F with H/d for Constant W/B = 5.0

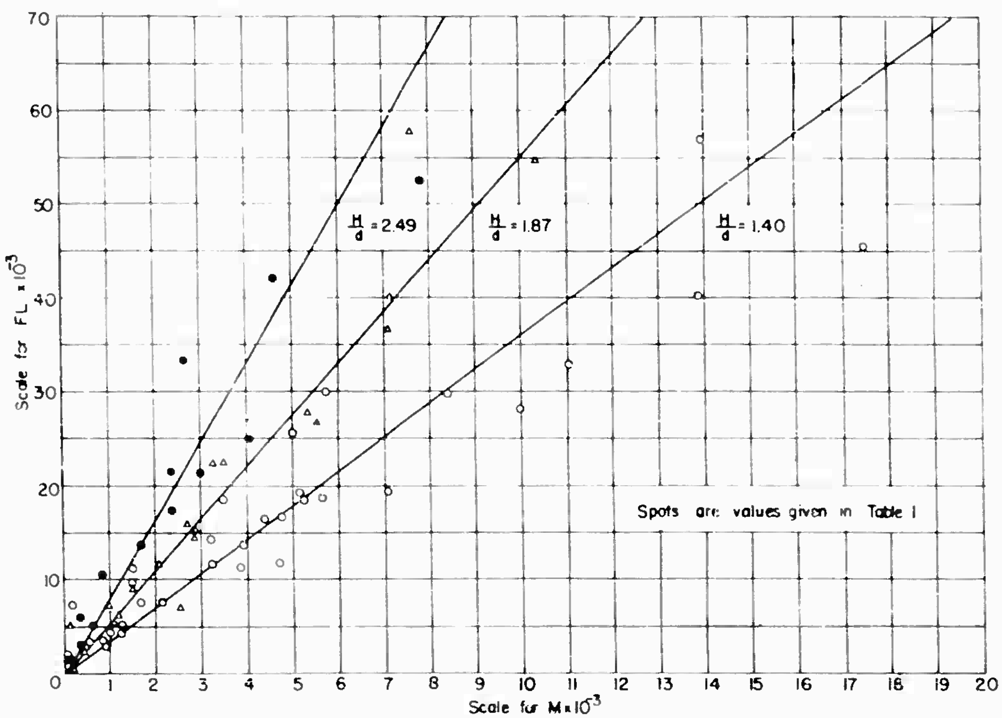

Figure 3 -- Graph of FL versus Yawing Moment for Constant Values of H/d

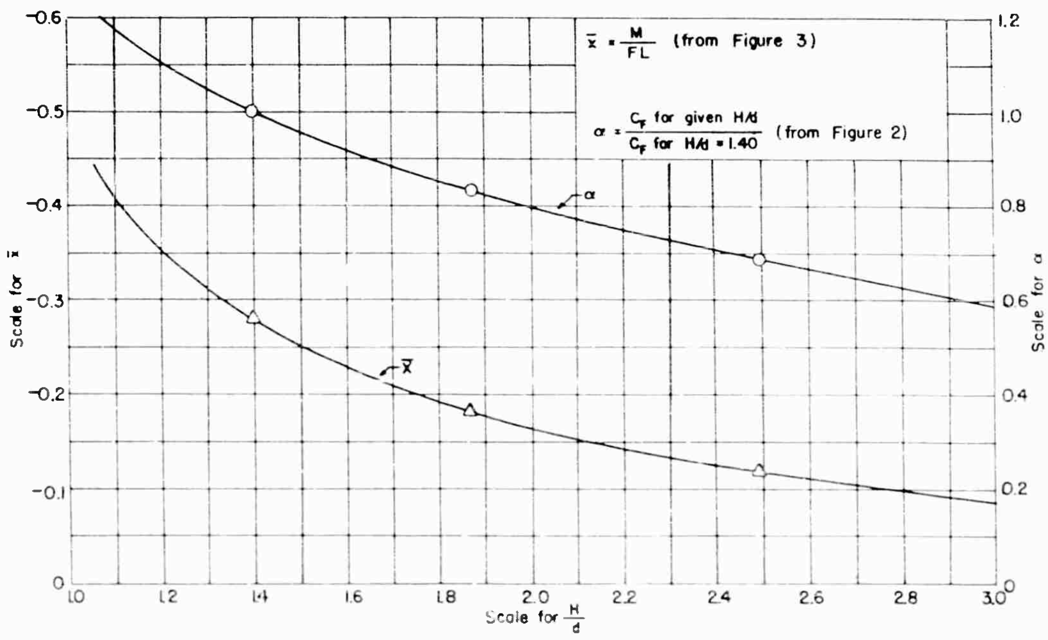

Figure 4 – Variation of $\bar{x}$ and x with H/d

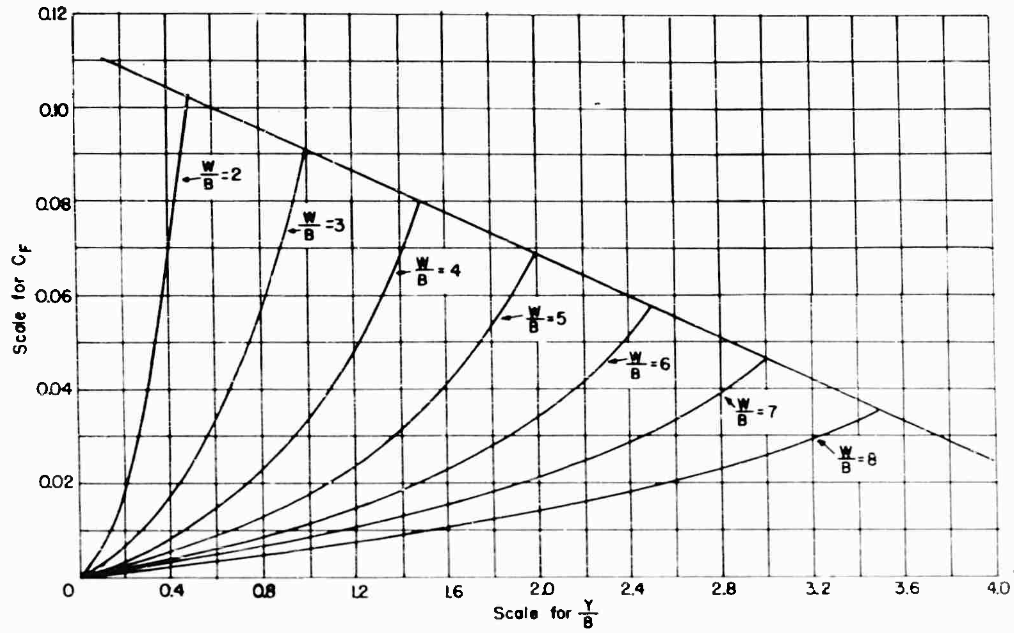

Figure 5 – Cross-Faired Values of C_F for $H/d = 1.40$ and Varying W/B

$$C_F = \frac{F}{\rho/2\,(Ld)\,v^2}$$

F = Lateral force in pounds

ρ = Fluid density

L = Ship length in feet

d = Ship draft in feet

v = Ship speed in feet per second

Y = Distance from channel centerline

B = Ship beam in feet

W = Channel width in feet

HANDLING QUALITY CRITERIA FOR SURFACE SHIPS

by

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Prepared for Presentation at the Meeting of the
Chesapeake Chapter of the Society of Naval Architects and Marine Engineers
to be held at the
David Taylor Model Basin, Washington 7, D.C.,
on 2 May 1959

ABSTRACT

The concept of definitive maneuvers is introduced as a means for providing numerical measures of handling qualities of surface ships which can ultimately lead to objective standards and specifications. Data derived from three basic types of definitive maneuvers, the spiral, overshoot, and turning circle, are presented to indicate the extent to which handling qualities differ among existing ship types that have been evaluated. Tentative criteria are proposed to serve as interim standards for selected qualities until more complete and systematic data become available. It is recommended that the effort to accumulate data be expanded to include numerical measures of a wider variety of handling qualities not only for existing ships but for research designs with near optimum stability and control characteristics.

INTRODUCTION

The subject of handling qualities of surface ships in its broadest sense deals with all types of responses of a given ship resulting from its own controls and external disturbances. The active controls of a ship consist primarily of its rudders and propellers, although some ships may be equipped with fins, tanks, or gyroscopes for actively stabilizing roll and pitch responses. The external disturbances arise from either environmental conditions such as wind, waves, and water currents or interaction effects due to passage within restricted channels or proximity to other ships.

It may be stated that the broad objective in the field of stability and control of ships is to achieve the best stability and maneuverability characteristics commensurate with other design requirements. It is not always obvious, however, what is categorically the "best" as in some other fields of naval architecture. Many of the previous attempts to define these elusive qualities have been highly subjective and wrapped up in the lore of the experienced ship operators. Furthermore, judgments are usually made after delivery and long term use of a ship rather than on the basis of predetermined goals.

It is evident, therefore, that there is a pressing need for a system of objective standards whereby desirable handling qualities for various ship types can be ascertained and rated both from the standpoint of the designer and operator. Since such standards represent the finite objectives to be achieved by the design process, the handling quality approach should serve as a foundation and actually precede all other approaches in the field of stability and control of ships.

The researchers have traditionally employed indices, derivatives, and hydrodynamic parameters to analyze stability and control characteristics. These methods may serve a very useful purpose as analytical tools and

undoubtedly contribute to the overall picture. Their weakness lies in their use as figures of merit since they usually lead to qualitative interpretations such as "acceptable" or "unacceptable." Furthermore, a profound knowledge of mathematics and systems analysis of the type found only among the highly specialized is required to fully understand the implications of such analyses. The operators, on the other hand, are concerned more intimately with ship behavior as it really exists in point of full-scale time and environmental forcing functions. The operators are the customers and must live with the ship long after the design has left the drafting table and research laboratory. Thus to establish an effective system for defining handling qualities, it is necessary to bridge this gap to enable a meeting of the minds of the researcher, designer, and operator.

The steps to be taken in dealing with the subject of handling qualities logically appear to fall in the following sequence:

1. Identification of significant handling qualities for various types of ships,

2. Formulation of test procedures or techniques to reveal these qualities in a quantitative or numerical sense,

3. Collection of handling quality data from full-scale trials and free-running-model tests of existing ships which are considered to be representative of the compromises that have been made between handling qualities and other design considerations,

4. Collection of handling quality data from model tests of research designs to establish the extent to which improvements can be realized over existing types,

5. Development of tentative handling quality criteria for assessing relative merit among existing and proposed designs, and

6. Establishment of handling quality specifications to be incorporated in the contractual negotiations for new ships.

Although the importance of establishing an effective system for rating handling qualities has been stressed, it should be borne in mind that this is only a first step toward achieving the ultimate refinement of the subject. Once it is clearly understood by all concerned what handling qualities are desirable and possible, the next obvious question is what must the designer do to realize these predetermined qualities? A well-rounded program on the stability and control of surface ships should include the following elements:

1. Studies of the handling qualities

2. Analytical studies of the equations of motion to determine effects of arbitrary changes in parameters or coefficients

3. Experimental studies to relate geometric variations to hydrodynamic forces and moments acting on bare hulls, control surfaces, and other appendages either singly or in combination

4. Theoretical studies of the basic mechanisms of the generation of hydrodynamic forces and moments acting on bodies moving through fluids

5. Analog computer or free-running model studies of complete configurations utilizing the data obtained in items 1 through 4.

A complete treatise covering all of the aspects of handling qualities of surface ships would be extremely lengthy and somewhat premature. The subject of this paper is confined, therefore, to handling qualities associated with horizontal plane motions of surface ships in essentially still water. This includes the ground covered by the terminology of steering (course-keeping) and maneuvering (course-changing). The primary purpose of this paper is to formulate a system for numerically defining the most significant of these handling qualities to enable a meeting of the minds of the researcher, designer, and operator with the ultimate objective of achieving superior surface ships from the standpoint of stability and control. To carry out this purpose, a brief history of the work related to this problem is given to provide some background. The concept of "definitive maneuvers" is then introduced as the basic framework for establishing a system for rating handling qualities. The particular maneuvers selected for this purpose are described and numerical measures obtained from model and full-scale tests employing these maneuvers are given for a number of commercial-and naval-type surface ships. Criteria are established to indicate good practices on the basis of those ship designs which have been sampled. Obviously, these are only tentative criteria for surface ships in general and will be subject to change as more detailed and progressive information becomes available. Finally, recommendations are made concerning future studies and trends that may tend to improve the state of development.

HISTORY

The subject of stability and control of ships and other watercraft dates back to ancient times. In fact it is as old as the first and most primitive of watercraft. The importance of being able to steer and maneuver watercraft must have been obvious even to prehistoric man. It is difficult, therefore, to understand why progress in this field has been so slow and

haphazard throughout the centuries. An excellent survey of the historical development of design "procedures" for maneuvering is given by Saunders in the forthcoming third volume of his book on "Hydrodynamics in Ship Design".[1] Consequently, the background given herein is confined to highlights which pertain to the development of handling quality criteria.

The formal aspects of the subject of stability and control of ships embrace some of the most difficult problems in hydromechanics. It is understandable, therefore, why the researchers have been attracted by this challenge and have concentrated on basic studies involving analyses of the coefficients of the equations of motion. At the other extreme, designers, experimenters, and operators have been left to their own devices and have relied upon empirical rules of sometimes obscure origin to obtain ships whose handling qualities were at least tolerable. As the result of this "conflict in interest" the problem of establishing common goals has never been resolved. A few attempts were made in the past to survey experienced operators to obtain their opinions as to what handling qualities they would like to see in their ships. These opinions have been extremely vague and widely divergent even among masters of sister ships. When the operators retaliated by asking the designers and experimenters what handling qualities they could supply, the answers were equally vague and noncommittal. On the basis of such experiences, it now appears that one of the first hurdles that must be overcome is the establishment of a common language to describe and precisely define handling qualities.

Most of the papers which give a modern treatment of the subject of stability and control of ships were issued after the year of 1940. It is of interest to examine a few of these in chronological order to determine the extent to which they coincide with the handling quality concepts outlined in this paper. One of the first papers which appears to be pertinent in this respect is Kempf's 1944 paper entitled, "Maneuvering Standards of Ships".[2] Here, the zig-zag maneuver is introduced as a method for defining a maneuvering "norm" for ships. A standard maneuver of this type was carried out with 75 different freighters. Both full-scale ships and models were used for these experiments. At first glance, this work appears to be directly applicable to the present concept since it attempts to provide a numerical standard of maneuverability for a given type of ship. It may be noted, however, that the yardstick employed for this purpose is the "period" of the particular zig-zag maneuver. It is believed that this period is an index which is of interest to people involved in making frequency response analyses rather than a quality which concerns the operator. Furthermore, this quantity is not definitive; a small period is not necessarily indicative of either good course-keeping or good course-changing ability.

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[1] References are listed on page 240.

However, the other data taken during the first half-cycle of the zig-zag maneuver such as time to reach execute, overshoot heading angle, and overshoot width of path are considered of more operational significance. These data could be quantitatively used as handling quality criteria. Unfortunately, the detailed data have not been published and have since been either lost or destroyed.

The paper "Turning and Course-Keeping Qualities," by Davidson and Schiff (1946)[3] appears by its title to be directly applicable to the present subject. In fact, its prologue suggests that numerical indices are needed so that both turning and ease of steering can be discussed in quantitative terms. The authors advocate that it is important to look into the experience of the past and inquire into what combinations have been realized in actual bodies, together with rudder sizes that have been needed. The main theme of the paper, however, is concerned with prediction techniques and tests to determine whether or not a ship is directionally stable rather than the handling qualities themselves.

The papers by Dieudonné[4] present another valuable tool which can be utilized in assessing handling qualities. Although, the spiral was first introducted as a maneuver which could be used to qualitatively determine whether or not a ship is directionally stable, it now appears that it can be quantitatively interpreted. The author in fact suggests that the results of such maneuvers could be used quantitatively to indicate the ease of steering a ship.

Recent papers by the Japanese, presumably dealing with steering and turning qualities of ships,[5,6] are concerned primarily with analysis and prediction techniques rather than the establishment of handling quality criteria.

Within the past three years, there has been a concerted effort directed toward the development of techniques and the accumulation of data for the purpose of establishing handling quality criteria for submarines. Naturally, this information is contained in classified reports. The concepts and techniques which have been developed are, in many cases, applicable to the surface ship problem, and in fact, were utilized in forming the underlying philosophy of this paper. Thus it can be said at the present time, that the approach to the problem is reasonably understood. The task that remains is to utilize this approach to collect sufficient data to support a system for rating the handling qualities of surface ships.

It is of interest to observe the progress made in the allied field of handling quality criteria for aircraft. A recent paper prepared by the North Atlantic Treaty Organization Advisory Group for Aeronautical Research makes the following statements concerning handling qualities of aircraft:[7]

"Between these early days and the early years of the forties, handling qualities were pretty generally subject to the judgment or whims of the individual designer or project pilot. The artistic aspect of configuration design was prevalent. Even today vestiges of this artistic approach are present in one or two companies in the United States. In the latter years of this period, an all-encompassing statement appeared in the U. S. Army Air Force designer's handbook that the stability and control characteristics should be satisfactory."

"The modern concept of handling qualities requirements began with the collection and codification of data that resulted in the publication of an NACA report by Gilruth in 1943. Issuance of Gilruth's report was followed shortly by the first set of Air Force requirements, C-1815."

"The basic questions that these specifications seek to detail is, "What will pilots tolerate"? To some degree and for certain requirements the specifications reflect the question "What do pilots like"? It has been a common misconception to interpret the specifications as design points rather than minimum tolerable points. This has been the source of poor characteristics for certain aircraft."

"It has been proposed that numbers be established for what is really desired, but generally this has not been done as yet."

It may be gleaned from the foregoing excerpts that progress with handling qualities of surface ships has been more or less paralleled by progress with handling qualities of aircraft. The aircraft people appear to have started attacking the problem earlier, but their present state of development is still far from being an exact process. The Society of Naval Architects and Marine Engineers took its first official step in the field of stability and control of ships by establishing the H-10 Controllability Panel in 1955. One of the primary functions of this panel is to encourage research and collect data on the handling qualities of surface ships.

CONCEPT OF DEFINITIVE MANEUVERS

The term "definitive maneuvers" has been adopted to describe a class of maneuvers designed solely to reveal objective or numerical measures of specific handling qualities. Some of these maneuvers may resemble operational maneuvers. It is highly desirable if this is the case since the numerical measures derived from the definitive maneuver will then have a more direct significance to the operator. There are two general types of definitive maneuvers, those which define inherent qualities of a ship resulting from its hydrodynamic design and are independent of the man or control mechanisms in the loop; and those which define qualities associated with the complete ship-control system and are dependent on the responses of the man, automatic control equipment, and control linkages in the loop.

Maneuvers which define inherent qualities are considered preferable because they directly provide specific numerical measures from a single maneuver of a given type. Also, these measures are indicative of the maximum potentialities of the ship system without qualification as to the efficiency of the operator in executing the maneuver. On the other hand, maneuvers which define qualities associated with the complete system are much more cumbersome and time consuming. Such a maneuver must be repeated many times with several operators, possessing varying degrees of skill, to furnish data which then must be statistically analyzed to obtain the desired figures of merit. Nevertheless, there are cases where the latter type of maneuver must be used if certain handling qualities are to be directly manifested.

The qualities associated with course-keeping are illustrative of a case where the statistical approach might be used. These qualities result from an interplay between the ship system consisting of the man or automatic control, the directional stability of the ship, the rudder effectiveness, and the control mechanism characteristics, and the external disturbances provided by currents, wind, and waves. Since the whole ship system is involved, appropriate numerical measures can be obtained only by conducting statistical-type course-keeping maneuvers.

Numerical measures pertaining to the inherent directional stability of a ship can be simply obtained by conducting a single spiral maneuver of the type attributable to Dieudonné.[4] Thus, if it is assumed that the ship with the best directional stability characteristics potentially will have the best course-keeping qualities, the numerical measures from the spiral maneuver can be used in lieu of those from the statistical course-keeping maneuver. Up to the present time, it has been necessary to make this assumption since most full-scale surface ships have not been available for properly conducted course-keeping tests. The only other alternative for providing course-keeping data would be to use simulator techniques similar to those used to evaluate performance of submarines. Unfortunately, neither hydrodynamic data nor well-developed techniques are available yet to support such studies with surface ships.

To gain a fuller appreciation of the concept and purpose of definitive maneuvers, it is helpful to temporarily forget the existence of other analytical methods and detailed approaches used to solve stability and control problems. If this is done it can be readily seen that the end product desired from stability and control studies points to those kinds of handling qualities that are of interest to the ship operators. Furthermore, the emphasis should be on treating these qualities in a quantitative sense if there is to be any hope of achieving progressive improvements on future ships. As a matter of orientation, it is desirable to consider descriptively the kinds of qualities of interest. The following is a list, which is by no means all-inclusive, of kinds of handling qualities that the operators should reasonably expect from a surface ship:

1. The ability to maintain course with a small amount of heading error, course error, and rudder activity

2. The ability to initiate a course change rapidly

3. The ability to check a course change rapidly with small overshoots in heading angle and width of path

4. The ability to execute an efficient steady-turning maneuver with small tactical diameter, advance, and transfer

5. The ability to accelerate and decelerate rapidly yet retaining good control

6. The ability to maneuver in and out of harbors ahead and astern at slow speeds without tug assistance.

The quantitative measures obtained from definitive maneuvers which are used to describe qualities of the kinds given in the foregoing list are discussed in the next section. It is pertinent to the concept of definitive maneuvers, however, that such numbers be expressed dimensionally in terms of real time and distance. In this manner, the numbers can be maintained within the perspective of the operators. In addition, they will serve as a better basis for specifications since they can be checked directly in acceptance trials. If it is desired, however, to utilize these numbers in analyses involving different-sized ships, the dimensional values can be converted into nondimensional ones by the use of appropriate normalizing factors.

On the basis of the preceding considerations, a given maneuver can be classified as a definitive maneuver if it has the following characteristics:

1. It can actually be performed by a full-scale ship and is not merely a laboratory or analytical response technique.

2. It has salient features which can be expressed as quantitative measures of specific handling qualities of the type that lead to objective standards and finally to specifications which must be met prior to the acceptance of a ship.

3. If possible, it should accomplish its purpose with a minimum of specialized instrumentation and without using a disproportionate amount of full-scale trial time.

DESCRIPTION OF SELECTED MANEUVERS

A wide variety of maneuvers have been used in the past as definitive maneuvers and others might conceivably be used in the future. Obviously, many of these maneuvers involve similar modes of performance and to this extent overlap each other in defining certain types of handling qualities. Consequently, in selecting standard definitive maneuvers, one can go from one extreme by considering too few maneuvers and perhaps overlooking some important handling qualities to the other extreme by utilizing too many and thus overburden trial schedules and produce excessive amounts of data. In the present stage of development of handling quality criteria, the conservative approach would be to select more instead of less than the required minimum number of representative maneuvers. In this manner, there would be less risk of overlooking some handling qualities that might become important in the future and the opportunity to conduct trials on a given ship may not again present itself. In any event, the number of maneuvers conducted on any given set of trials will be compounded by the range of speeds (forward and backing) as well as other pertinent conditions. A thorough coverage of operational conditions should be considered in establishing handling quality criteria. It is unwise at this stage of development to place too much credence on handling qualities of a single type. This point has been confirmed by recent experiences with naval-type surface ships. For years, the maneuverability of naval ships was evaluated solely on basis of steady-turning tests. However, within the last few years it was found that some of the ships which had excellent turning characteristics had poor and, at least in one case, unacceptable directional stability characteristics. As the result of these findings, spiral tests to define directional stability characteristics have now become as standard with naval ships as the traditional turning tests.

The development of facilities, instrumentation, and techniques which are necessary for detailed treatment of the subject of stability and control of surface ships has been relatively slow. In addition, full-scale surface ships have been made available for only limited programs to evaluate maneuverability. Consequently, whatever data are available have been obtained from essentially three types of definitive maneuvers, spirals, overshoots, and turning circles. Each of these three types of maneuvers are discussed in terms of the purpose of the maneuver, the procedure followed in carrying out the maneuver, the numerical measures derived from it, and the significance of the numerical measures.

SPIRALS

The spiral is a definitive maneuver which is intended to provide quanitative measures of the inherent directional stability characteristics of a ship. These characteristics can be used to impute course-keeping potentialities. The maneuver can be conducted in a variety of ways with full-scale

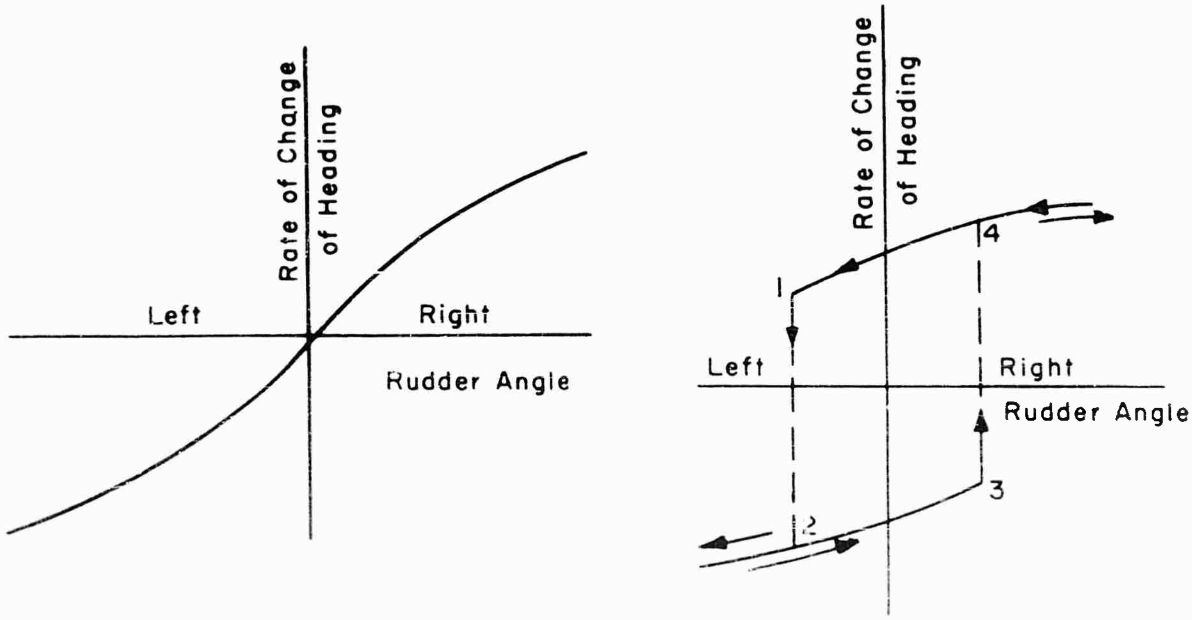
ships, free-running models, and analog computers utilizing hydrodynamic force and moment data derived from captive model tests. An attractive feature of the maneuver for full-scale tests is that it can usually be carried out with the ship's own instrumentation. The basic maneuver, which can be carried out when sea room is not at a premium, is conducted as follows:

1. The propeller speed is adjusted to an rpm corresponding to a predetermined speed (either ahead or astern). Once a steady rpm is achieved, the throttle settings are not changed for the balance of the maneuver.

2. The rudder is manipulated as necessary until a "practically" straight course has been obtained and held for one minute.

3. The rudder is then deflected to about 15 degrees right and held until the rate of change of heading as indicated by the gyro compass and a stop-watch remains constant for one minute. The rudder angle is then decreased by 5 degrees and held again until the rate of change of heading remains constant for one minute. The procedure is repeated until the rudder has covered a range of from 15 degrees on one side to 15 degrees on the other side and back again to 20 degrees on the first side. For 5 degrees on either side of zero or neutral rudder angle, the intervals are taken in one degree steps.

The numerical measures obtained from the spiral maneuver are the steady rates of change of heading versus rudder angles. A plot of these variables is indicative of the inherent characteristics of the ship. If the plot is a single continuous curve going from right rudder to left rudder, as shown in Figure 1a, the ship is said to be directionally stable. If, however, the plot consists of two branches joined together to form a "hysteresis" loop, as shown in Figure 1b, the ship is said to be directionally unstable. In addition, the size of the loop (height and width) can be used as a numerical measure of the degree of instability; the larger the loop, the more unstable the ship. The width of the loop is also a fairly direct indication of probable course-keeping ability since it defines the envelope of rudder angles which must be employed to keep the ship from swinging from port to starboard. Unfortunately, the spiral technique as presently used does not define the degree of stability for stable ships. The slope of the rate curve at the origin seems to be indicative of degree of stability for directionally stable ships. Also, the time required for the turning rate to decrease to zero when the rudder is returned to zero or neutral angle may provide a numerical measure of degree of stability. Further analysis of these techniques is required to establish these relationships, however, and it may develop that a supplementary definitive maneuver may be needed in the case of directionally stable ships.



(a) Stable Ship (b) Unstable Ship

Figure 1 - Typical Curves from Spiral Maneuvers

OVERSHOOT

The overshoot is a definitive maneuver which is intended to provide quantitative measures of the inherent effectiveness of the rudder in making changes in heading or width of path. The kinds of handling qualities revealed by this maneuver are typified by the ability to initiate course changes and ability to check course changes during transient maneuvers. The maneuver can be conducted with full-scale ships, free-running models, and analog computers. The numerical measures pertaining to the heading changes can be obtained with the ship's own instrumentation. Numerical measures associated with width of path, however, will require either much more elaborate equipment than is generally available for most ships or testing on a range with triangulation facilities.

The overshoot maneuver is shown diagrammatically in Figure 2. It can be seen that if the maneuver is continued through several cycles it results in the well-known zig-zag maneuver. A typical procedure for conducting overshoot tests is as follows:

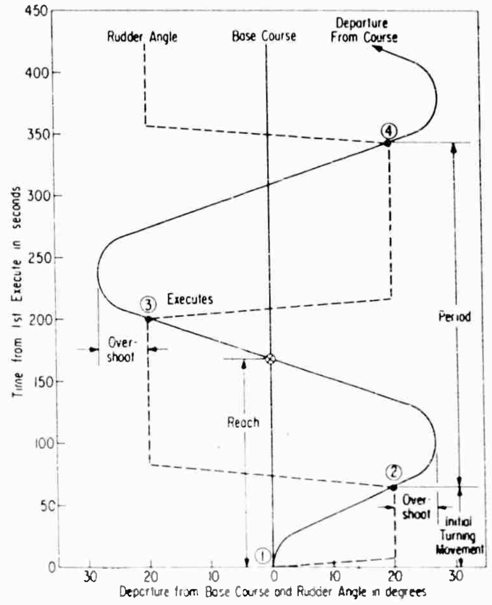


Figure 2 - Diagram of Overshoot Maneuver

1. The propeller speed is adjusted to an rpm corresponding to a predetermined speed and when a steady rpm is achieved, the throttle settings are not changed for the balance of the maneuver.

2. The rudder is manipulated as necessary until a "practically" straight course has been obtained and held for one minute.

3. After steady conditions on straight course have been established, the initial heading shown on the ships gyro compass is noted. The rudder is then deflected at maximum rate to a predetermined angle, say 20 degrees, and held until a predetermined execute change of heading angle, say 20 degrees, is reached.

4. At this point, the rudder is deflected at maximum rate to an opposite (checking) angle of 20 degrees and held until the ship passes through its initial course.

5. If a zig-zag is to be completed, the maneuver is continued until a second execute of 20 degrees to the other side is reached. Whereupon, the rudder is again deflected rapidly to an angle of 20 degrees in the first direction. This cycle is repeated through 3rd and 4th executes and so on.

The primary numerical measures obtained from the overshoot maneuver are the time to reach execute change of heading angle, overshoot heading angle, and overshoot width of path. The zig-zag maneuver provides the additional measures of reach and period which are perhaps more significant for frequency response analyses than establishment of handling qualities.

The time to reach execute is a direct numerical measure of ability to rapidly initiate changes in course. The heading and path-width overshoots are measures of course-checking ability and are indicative of the amount of anticipation and latitude of error that the helmsman is permitted if he is to remain within tolerable limits of the maneuver.

TURNING CIRCLES

The turning circle is a definitive maneuver which is intended to provide quantitative measures of the effectiveness of the rudder in producing steady-turning characteristics. The turning circle is the oldest, most familiar, and most widely used of the definitive maneuvers. The handling qualities defined by this maneuver are generally considered to be more important to naval than most sea-going merchant ship applications. The maneuver can be conducted with full-scale ships, free-running models, and ultimately with analog computers. As with the other maneuvers, some of the desired numerical measures can be obtained with the ship's own instrumentation in open sea. However measures pertaining to path data will require either much more elaborate ship-borne equipment or testing on a range with triangulation facilities.

Although the turning circle maneuver is familiar to most naval architects, it is shown diagrammatically in Figure 3 for purposes of completeness. The standard procedure for the conduct of such maneuvers is as follows:

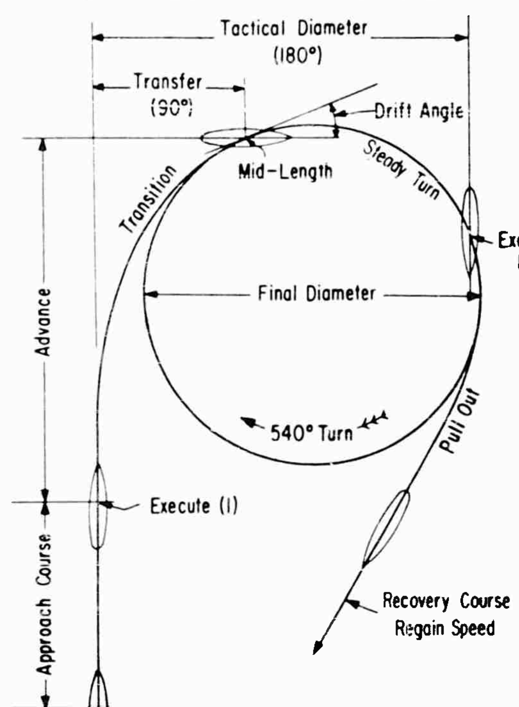


Figure 3 - Diagram of Turning Circle Maneuver

1. The propeller speed is adjusted to an rpm corresponding to a predetermined speed and when a steady rpm is achieved, the throttle settings are not changed for the balance of the maneuver.

2. The rudder is manipulated as necessary until a "practically" straight course has been obtained and held for one minute.

3. After steady conditions on straight course have been established, the initial heading on the ship's gyro compass is noted. The rudder is then laid to a predetermined angle, say 35 degrees, and held until a change of heading of generally at least 540 degrees has occurred at which point the maneuver is terminated.

The numerical measures obtained from the turning circle tests are the tactical and steady-turning diameter, advance, transfer, times to change heading 90 and 180 degrees, and loss of speed in turning. All of these measures should be taken into consideration in defining handling qualities associated with this type of maneuver.

NUMERICAL MEASURES FOR VARIOUS SHIPS

It has not been common practice in this country to carry out either full-scale maneuvering trials or model tests to evaluate handling qualities of commercial ship types.[8] In fact, it has only been in recent years that naval ship types have been tested to evaluate handling qualities other than those associated with turning circles. Consequently, there is only a limited amount of such data in existence. Furthermore, due to classification restrictions, only a small amount of the existing data is available for general publication. It is hoped, therefore, that enough interest in the problem will be generated to encourage ship owners to carry out the necessary tests with existing and new ships and thus contribute to the general fund of data on handling qualities of surface ships.

The geometrical characteristics and numerical measures obtained from definitive maneuvers of unclassified ships that have been tested by the David Taylor Model Basin are given in Tables 1 and 2, respectively. It may be noted that all values are given dimensionally to preserve their significance to the operators and thus comply with the concept outlined earlier in this paper. Sufficient data are given, however, to allow those who prefer to make an analysis on basis of nondimensional coefficients or ratios to do so. The designations A, B, C, etc., indicate the various different ships. Upper case letters are used when the data have been obtained from full-scale trials. Lower case letters are used when the data have been derived from free-running model tests.

TABLE 1

Geometrical Characteristics

Designation	Description	Length ft	Beam ft	Draft ft	Trim ft	Displacement ton	Rudder Area sq/ft
A	SS-SR*	435	63	22.75	3.5 aft	12,100	170
B	SS-SR	486	72	25.5	6.5 aft	15,100	292
C	SS-SR	475	72	18.63	6.1 aft	10,230	244
D	TS-SR	543	75	31.0	0 aft	24,275	314
E	TS-SR	525	75	26.25	1.0 aft	18,845	270
F	TS-SR	640	86	18.87	8.75 aft	19,000	392
G	TS-TR	500	82	17.0	4.0 aft	10,750	247
H	TS-TR and Sk	500	82	17.0	4.0 aft	10,750	247

Note: SS - Single Screw
SR - Single Rudder
TS - Twin Screw
TR - Twin Rudder
Sk - Skeg

TABLE 2

Numerical Measures from Definitive Maneuvers

Designation	Approach Speed knots	Spirals: Height of Loop degrees per second	Spirals: Width of Loop degrees	Overshoots: Time to Reach Execute seconds	Overshoots: Overshoot Heading Angles degrees	Overshoots: Reach seconds	Overshoots: Period seconds	Turning Circles: Tactical Diameter yards	Turning Circles: Advance yards	Turning Circles: Transfer yards	Turning Circles: Time to Reach 90 degrees seconds	Turning Circles: Time to Reach 180 degrees seconds	Turning Circles: Speed Remaining After 180 degrees knots
A	7.5	0	0	70• R	3.0	165	295						
	7.5	-	-	68• L	3.5	172	295						
	7.5	-	-	75 R	5.0	185	345						
	7.5	-	-	76 L	5.5	185	345						
	15.0	0	0	40• R	4.0	97	165						
	15.0	-	-	40• L	4.0	98	168						
	15.0	-	-	44 R	7.5	108	193						
	15.0	-	-	43 L	8.5	113	207						
•	7.5	0	0	71•	4.0	183	309						
	7.5	-	-	80••	6.0	190	310						
	7.5	-	-	71	7.5	133	333						
	15.0	0	0	37•	5.0	103	175						
	15.0	-	-	38••	6.5	104	179						
	15.0	-	-	39	9.0	103	184						
B	8.0	0	0	65•	3.0	160	260						
	8.0	-	-	70	5.0	185	315						
	12.0	-	-										
	12.0	-	-										
	16.0	0	0	37•	4.0	90	137						
	16.0	-	-	40	5.0	88	150						
	20.0	-	-										
	20.0	-	-										
b	14.0	0	0	35•	3.5	110							
	14.0	-	-	[illegible]8	6.0	83	121						
C	10.0	0	0	55• R	4.5	125	195						
	10.0	-	-	50• L	4.0	12[illegible]	206						
	10.0	-	-	50 R	7.0	128	227						
	10.0	-	-	47 L	7.0	123	229	695 R	565	320	119	234	6.5
	12.0	-	-	[illegible]7• R	4.0	77	135	630 L	565	285	114	217	6.5
	12.0	-	-	[illegible]7• L	5.5	79	131	735 R	600	340	93	178	10.9
	17.0	0	0	34 R	10.0	91	158	705 L	610	315	89	169	11.1
	17.0	-	-	36 L	9.5	92	155	770 R	635	355	76	142	13.4
	21.5	-	-					780 L	660	340	77	143	14.3
	21.5	-	-										
c	10.0	0	0	45• R	4.0	123	215						
	10.0	-	-	47••	6.8	130	229						
	10.0	-	-	47	7.5	127	239						
	17.0	0	0	34•	5.0	89	148						
	17.0	-	-	43••	7.0	99	113						
	17.0	-	-	38	8.5	97	111						

Note: • 10-10 zig-zag
•• 15-15 zig-zag

TABLE 2 (continued)

Numerical Measures from Definitive Maneuvers

Designation	Approach Speed knots	Spirals: Height of Loop degrees per second	Spirals: Width of Loop degrees	Overshoots: Time to Reach Execute seconds	Overshoots: Overshoot Heading Angles degrees	Overshoots: Reach seconds	Overshoots: Period seconds	Turning Circles: Tactical Diameter yards	Turning Circles: Advance yards	Turning Circles: Transfer yards	Turning Circles: Time to Reach 90 degrees seconds	Turning Circles: Time to Reach 180 degrees seconds	Turning Circles: Speed Remaining After 180 degrees knots
D	10.0	-	-					885	682	445	182	352	6.5
	15.0	-	-					875	655	465	122	235	9.7
	18.0	-	-					865	675	504	102	200	11.4
E	8.0	0.22	5.0	96*	4.0	238	535						
	8.0	-	-	100	9.5	275	493						
	16.0	0.58	5.0	60*	7.0	185	390						
	16.0	-	-	53 R	11.5	155	285						
	16.0	-	-	56 L	14.0	165	310						
e	8.0	0.24	6.4	93*	4.0	278	-						
	8.0	-	-	90**	5.0	250	-						
	8.0	-	-	93	7.0	252	-						
	16.0	0.42	2.5	53	7.0	169	-						
F	10.0	0.36	9.0	67	10.0	164	308						
	12.0	-	-	-	-	-	-						
	17.0	0.34	5.0	49	11.0	119	210						
	20.0	-	-	-	-	-	-						
f	10.0	0.14	3.0	77	6.5	177	308						
	17.0	0.22	4.5	56	10.5	135	420						
G	7.0	0.82	18+	71	28.0	261	-						
g	7.0	0.72	13.0	-	-	-	-						
H	7.0	0.26	4.0	97	11.0	236	-						
	7.5	-	-	77	10.0	195	344						
	15.0	-	-	43	20.0	123	209						
h	7.0	0.12	2.0	-	-	-	-						

Note: * 10-10 zig-zag
** 15-15 zig-zag

The numerical measures in Table 2 are derived from the spiral, overshoot, and turning circle maneuvers. The measures associated with the spiral maneuver are the maximum variation of steady heading rate at zero or neutral angle (height of hysteresis loop) and maximum variation of rudder angle at zero steady heading rate (width of loop). For directionally stable ships, both of these quantities become zero and beyond this point there is no further indication of "degree" of stability. The overshoot maneuvers are essentially zig-zag maneuvers conducted either with rudder angles of ± 20 degrees and execute heading angles of ± 20 degrees or rudder angles of ± 10 degrees and execute heading angles of ± 10 degrees. The former are considered to be more preferable for defining course-changing ability; the latter are directly comparable with Kempf's data. The measures taken during the first half cycle of the maneuver, namely time to reach execute and overshoot heading angle are considered to be most significant. However, the reach which is the time to complete the first half cycle of the heading trajectory and the period which is the time to complete succeeding whole cycles are also listed for comparative purposes. The numerical measures taken from the turning circle maneuver are the tactical diameter, advance, transfer, time to reach 90 degrees change of heading, time to reach 180 degrees change of heading, and loss of speed after 180 degrees change of heading. For any of the foregoing measures, the best performance is characterized by the lowest value. However, some of the qualities have a tendency to be conflicting and, therefore, it may not be possible for a given ship to have all of the lowest numbers among a comparable group of ships.

It is of interest to examine the range of pertinent handling qualities among the existing ship types that have been evaluated. This can be accomplished with graphs showing the individual numerical measures. Data available from other naval ships are included to make the survey as representative as possible. Since these data are classified, they are not identified or related to specific ships. The values for all ships considered have been corrected to correspond to a 500-foot version of each design to retain the dimensional characteristics without becoming involved in other ramifications. These values can be interpreted as applying with reasonable accuracy to ships between 300 and 700 feet in length.

The numerical measures from spiral maneuvers are presented by the bargraphs in Figure 4. To simplify the graphs, the height or width of the hysteresis loop for each ship was averaged over a range of ship speeds between 5 and 20 knots. The bars are constructed as percentages of the total number of the ships in the survey. It may be noted that more than one-half of the ships are directionally stable. Even though they are in active service, most of the remaining ships have characteristics which are not considered desirable on the basis of the standards that are being established. In a few isolated cases, the degree of directional instability is so great that the ships are difficult and hazardous to maneuver.

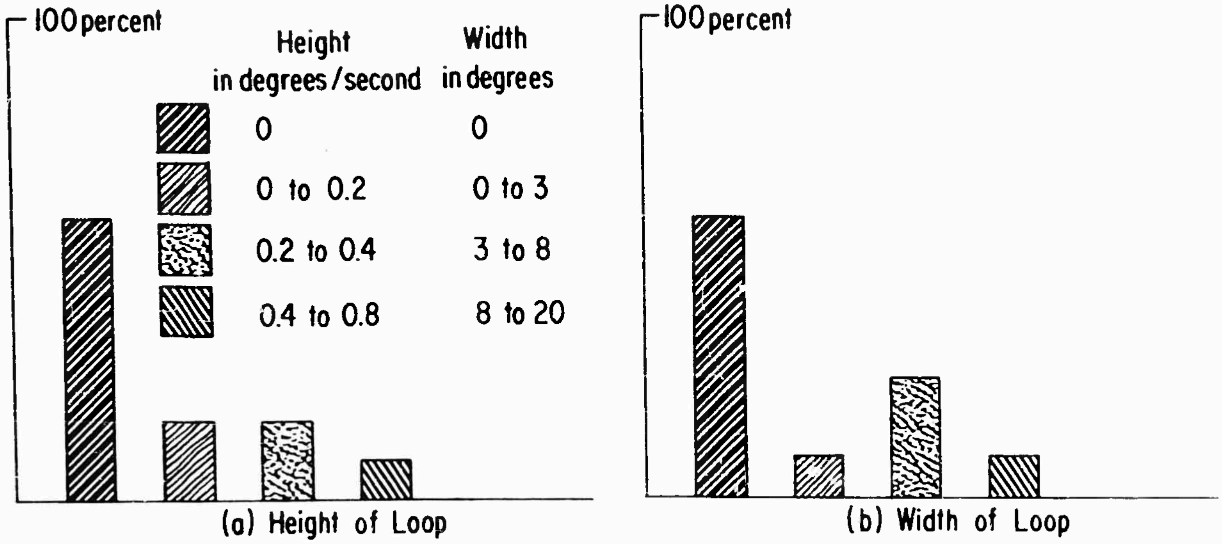


Figure 4 - Bargraph of Numerical Measures from Spiral Maneuvers

As mentioned earlier, it is not possible to assess the status of maneuverability of commercial ships. It may be reasonably inferred, however, that they will generally have somewhat poorer handling qualities than comparable naval ships since their operational requirements are not usually as severe.

Ships which have no loop as the result of spiral maneuvers should have good course-keeping ability. Those with a wide loop can be expected to require an excessive use of the rudder with attendant wear and tear on the steering machinery and fatigue of the helmsman. In addition, the excessive rudder travel will probably result in an increase in resistance and consequent increase in fuel consumption. It is believed that the foregoing predictions can be reasonably inferred from the spiral results. It would be desirable, however, to have course-keeping data for corroboration.

To illustrate the consequences of a high degree of directional instability, the case of one of the ships studied, a twin-rudder naval auxiliary, may be considered. On the basis of model turning tests, the ship was expected to have very good turning characteristics. Since it was not standard procedure at the time, model maneuvering tests were not conducted. Unfortunately after the ship was built, it exhibited an unhappy facility for running aground when negotiating a channel which led to the building yard. Upon delivery to the Navy, it became obvious that the maneuvering characteristics of the ship had to be improved. The results of full-scale spiral tests indicated a hysteresis loop (for a 500 ship) which was 0.82 degrees per second high and over 18 degrees wide. On the way to the open sea area, it was noted that the helmsman habitually used ± 15 degrees rudder angle for normal course-keeping in smooth seas. These large rudder angles may have been influenced to some extent by the lack of physical exertion required to spin the wheel. There is no doubt, however, that at least ± 10 degrees rudder angle was necessary to maintain course.

As a result of model tests, an enlarged skeg was designed and welded to the ship. The loop was reduced to 0.26 degrees per second high and 4.0 degrees wide. It was not practicable to install a large enough skeg to completely eliminate the loop, however, the resulting performance was sufficient to meet the operational requirements of the ship.

The numerical measures derived from overshoot maneuvers of the various ships surveyed, including the classified naval ships, are compared in Figures 5 and 6. The values in the figures have been adjusted to correspond to 500-foot ships. All values have been obtained from a 20-20 overshoot maneuver and consist of the time to reach 20 degrees execute change of heading using 20 degrees rudder angle and overshoot angle using a rudder angle of 20 degrees to check the swing.

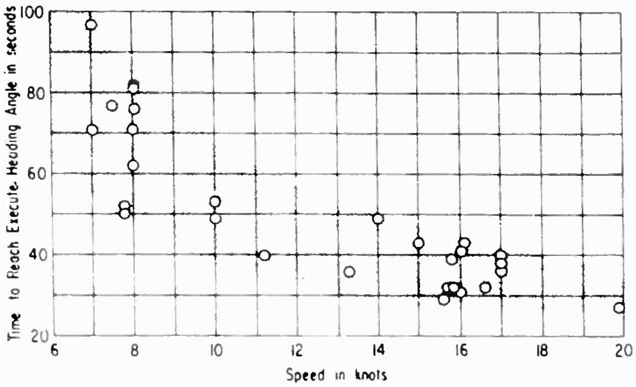


Figure 5 - Times to Reach Execute from 20-20 Overshoot Maneuvers

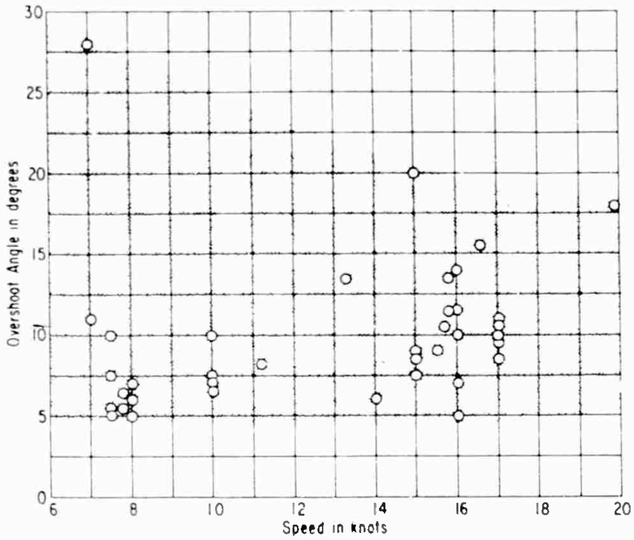


Figure 6 - Overshoot Angles from 20-20 Overshoot Maneuvers

The envelopes surrounding the spots in Figures 5 and 6 exhibit considerable spread among existing ships both in time to reach execute and overshoot angle. This suggests room for significant improvements in these respects and should serve as an incentive and challenge to the designers.

The rapidity with which a turn can be initiated (time to reach execute) appears to be determined primarily by the effectiveness of the rudder in providing turning moment to the ship. On the basis of an examination of the other characteristics of the ships corresponding to the spots on Figure 5, the directional stability does not appear to influence the time to reach execute. On the other hand, the ships with greater rudder effectiveness (those with rudders in the propeller race) appear to group themselves near the lower bound of the envelope curve for time to reach execute.

Figures 7 and 8 illustrate the effects of control effectiveness and directional stability on overshoot characteristics. Figure 7 shows trajectories from a 20-20 overshoot for two comparable naval auxiliaries. One of these ships is a twin-screw single-rudder type. The other is a single-screw single-rudder type with its rudder in the propeller slipstream. The latter is about 40 feet shorter but this difference in length is not considered significant. Although the twin-screw ship is somewhat unstable, this should not affect the comparison in regard to time to reach execute. The single-screw ship reaches 20 degrees execute, (2) in Figure 7, in 42 seconds whereas the twin-screw ship takes 56 seconds.

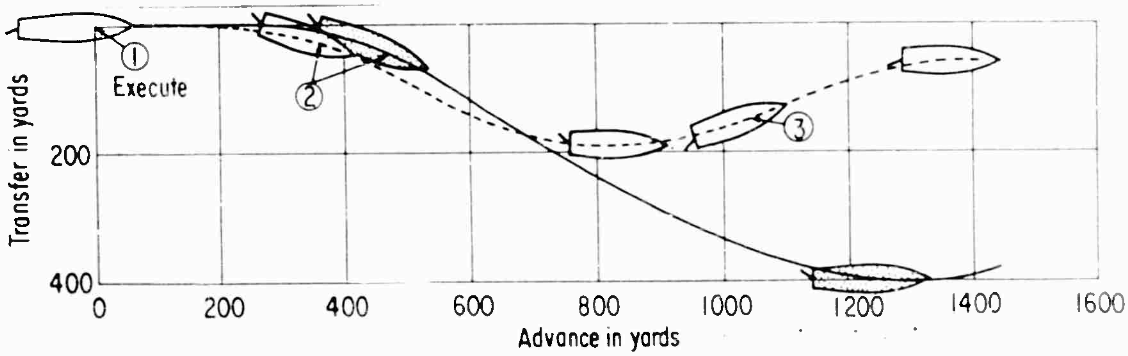


Figure 7 - Overshoot Trajectories of Two Different Types of Naval Auxiliaries

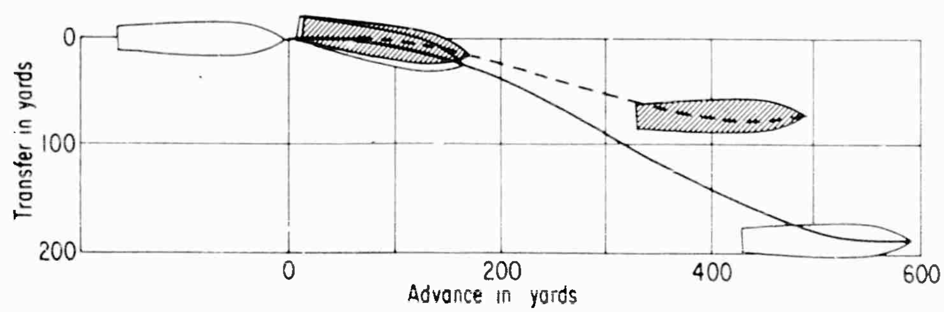


Figure 8 - Overshoot Trajectories of Twin-Screw Twin Rudder Naval Auxiliary with and without Centerline Skeg

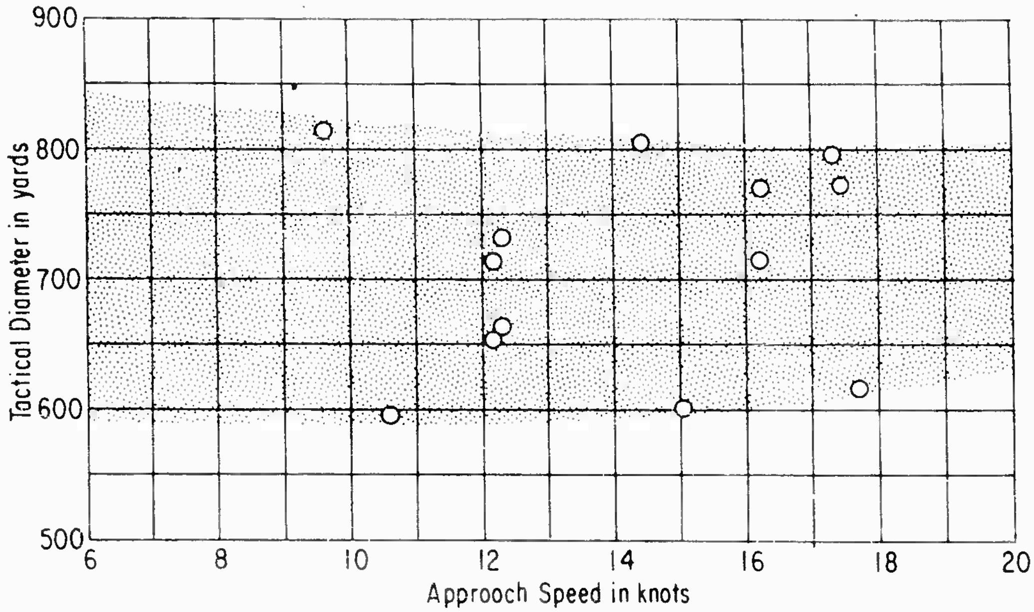


Figure 9 - Tactical Diameters from Turning Circle Maneuvers with 35 Degrees Rudder

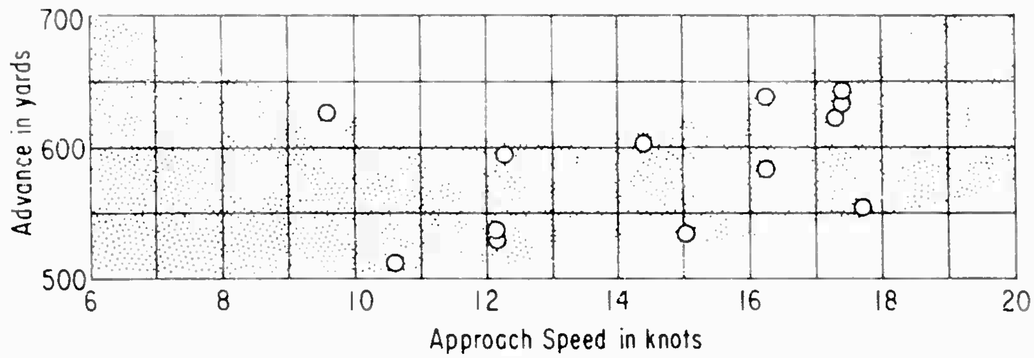


Figure 10 - Advances from Turning Circle Maneuvers with 35 Degrees Rudder

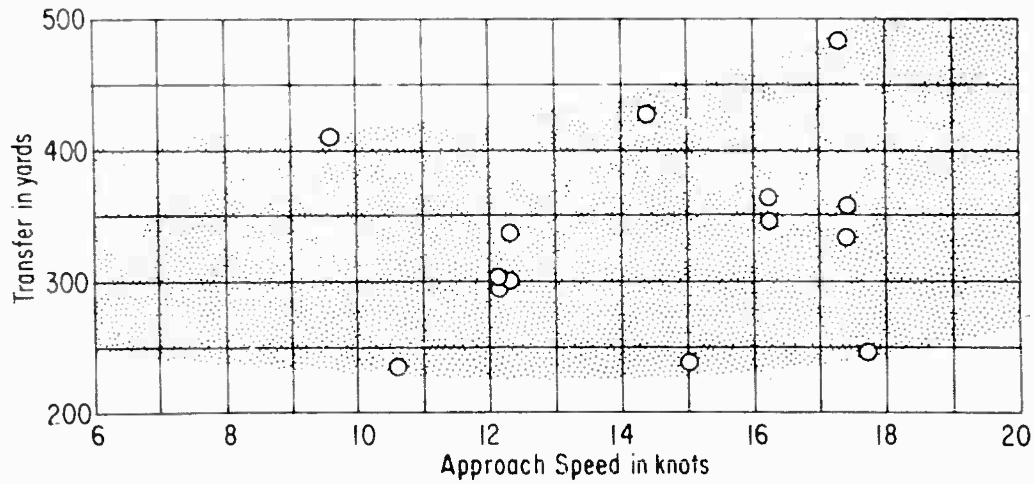


Figure 11 - Transfers from Turning Circle Maneuvers with 35 Degrees Rudder

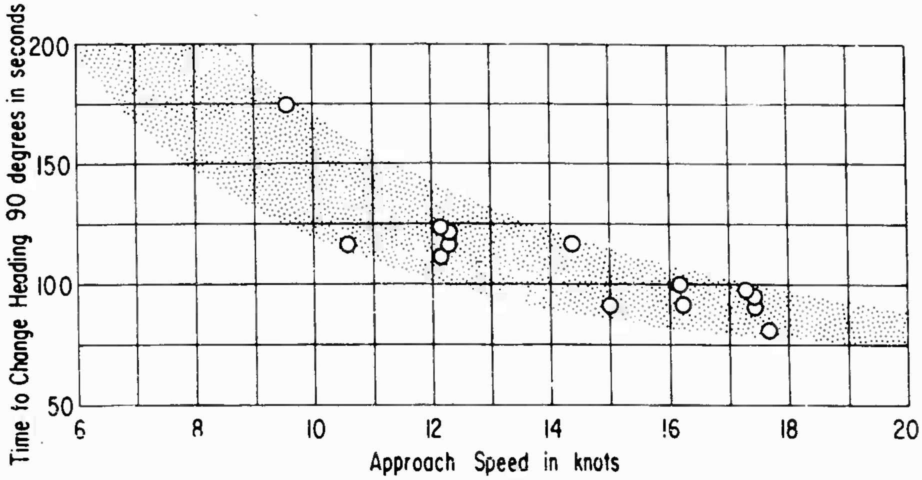


Figure 12 - Times to Change Heading 90 Degrees with 35 Degrees Rudder

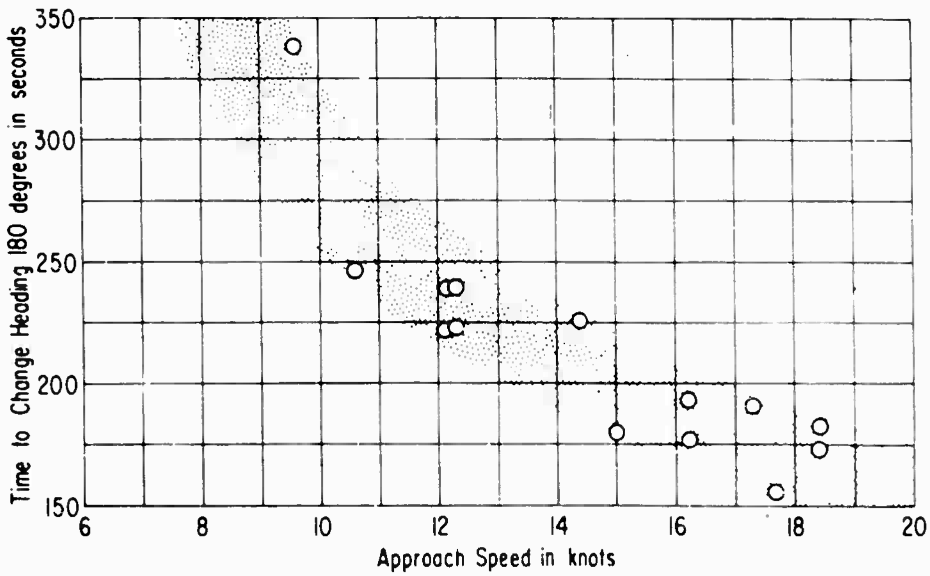


Figure 13 - Times to Change Heading 180 Degrees with 35 Degrees Rudder

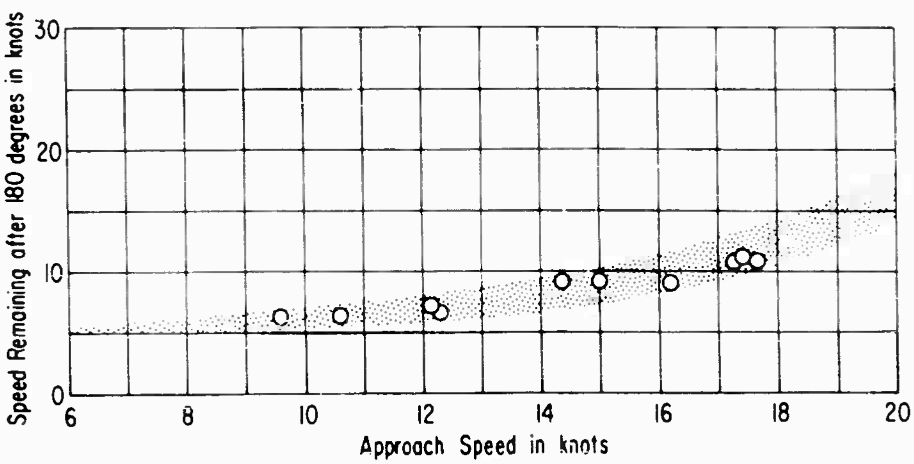


Figure 14 - Speeds Remaining after 180 Degrees Change in Heading from Turning Circle Maneuvers with 35 Degrees Rudder

The overshoot angles shown by Figure 6 appear to be affected both by directional stability and rudder effectiveness. It is difficult to say which of these factors has the strongest influence but it was noted that the excessively large overshoot angles were always obtained with the ships that had a high degree of directional instability. For example, in the case of the twin-rudder naval auxiliary whose directional instability was markedly decreased by addition of a skeg but whose rudder effectiveness was essentially unaltered, the overshoot angle was reduced from about 28 to 11 degrees.

As mentioned earlier, there are insufficient data available on overshoot width of path because of inadequate facilities for tracking. The width of path is of importance to the ship handler who is concerned with the path swept by the ends of the ship in checking course changes. This is particularly true in restricted waters and may mean the difference between damaging the ship or not. It is hoped that data of this type will be in more abundance after the new Maneuvering and Seakeeping Facilities at the Taylor Model Basin are put into operation. It is of interest at this time, however, to examine the effects of excessive directional instability on overshoot width of path for the case of the naval auxiliary mentioned in the preceding paragraph. Figure 8 depicts the results of overshoot maneuvers carried out with free-running models of the alternative designs using an execute change in heading of about 10 degrees with rudder angles ± 35 degrees. It may be seen that the overshoot width of path was reduced from about 175 to 65 yards by addition of the skeg. A similar overshoot test utilizing ± 20 degrees rudder was attempted with the highly directionally unstable ship but could not be completed within the width of the basin.

The advantage of realizing small overshoots can also be seen by reference to Figure 7. In this case, the total width of path changes from 400 to 190 yards for the comparable ships performing the same maneuver even though differences in directional stability are not too great.

The numerical measures from turning-circle maneuvers of the various ships surveyed are shown in Figures 9 through 14. Again, the comparisons are made on the basis of 500-foot ships. These figures demonstrate that, although turning circles have been studied more intensively than other maneuvering characteristics there is still a wide spread in turning performance among existing ship types. At first reaction, it appears that this can be explained on the basis of the different operational requirements for the various ships. The supposition may be made, for example, that the turning performance was sacrificed for the ships that must have excellent course-keeping ability. An examination of the various handling qualities among the ships surveyed does not support this contention. In fact, some of the tightest turning ships are directionally stable, and therefore, should have excellent course-keeping qualities as well. Conversely, some of the ships with the largest tactical diameters are directionally unstable and should exhibit poor course-keeping qualities.

In the past, the tactical diameter has been emphasized as the primary numerical measure of the effectiveness of a ship in turning circle maneuvers. It has been at least tacitly assumed that once the designer has exercised latitude in favor of a given tactical diameter, the values of the resulting quantities such as advance, transfer, times for heading changes, and speed remaining after 180 degrees are inevitable.[9] It is the philosophy of this paper to point out where ultimate refinements are possible rather than to compress the data into a rigid mold. Consequently, it is advocated that each of these numerical measures be scrutinized to see what improvements can be made in each without significantly affecting the others. For example, if a comparison is made on the basis of equal tactical diameter, it can be readily seen that among the ships surveyed there is a substantial spread in the values of advance, times for heading changes, and speed remaining after turning. Thus, there is evidence that the designer has some control over all of these qualities.

TENTATIVE HANDLING QUALITY CRITERIA

The numerical measures obtained from definitive maneuvers which have been presented herein constitute a relatively small sample of the handling qualities of existing ship types. Furthermore, the preponderant number of naval ships which, of a necessity, were included in the survey may affect interpretations when applied to merchant ship types. There is always a reluctance to make definite commitments or propose finite numbers, especially when a field of endeavor is in the formative stages. Nevertheless, some attempt should be made at this point to establish tentative criteria at least on those kinds of handling qualities covered by this paper. This may at least have the effect of familiarizing the profession with the use of the proposed rating system so that objective standards and specifications may emerge in the not-too-distant future.

It is fully realized that there are definite limitations and drawbacks to establishing criteria from insufficient data. It is hoped, however, that the tentative criteria will not be used too rigorously at this time as specifications or design objectives but rather as guides to good practices. In general, the tentative criteria which are proposed are pessimistic in the sense that it should be possible to do better when more detailed knowledge on the stability and control of ships becomes available. They may be optimistic, in the sense that they may not be fully realized with all ship types especially where the governing factors lie in other design considerations. In all cases, however, they should serve as guides for determining whether the price to be paid for achieving each and every number is reasonable in terms of the overall design.

For purposes of emphasizing the distinct modes of performance, the tentative criteria are grouped into those pertaining primarily to steering, maneuvering, and turning.

STEERING

In absence of adequate data from course-keeping maneuvers which could provide numerical measures of rudder activity, heading-angle deviation, and path deviation while maintaining course under specified environmental conditions, spiral maneuvers may be employed to provide reasonable measures from which steering qualities may be inferred. Complete elimination of the loop from the spiral is advocated in all cases to obtain a ship which is inherently directionally stable and tends to return to a straight path after a disturbance. The rudder angle is thus needed only to ensure that the path followed is on the desired course. As pointed out earlier, there is a unique turning velocity associated with any given rudder angle for stable ships whereas for unstable ships the direction the ship will turn is unpredictable within the bounds of the loop.

If it is not practicable to eliminate the loop entirely, every effort should be made to minimize both the height and width of the loop by suitable design of rudders and stabilizing surfaces. Any new design having a loop height exceeding 0.2 degrees per second (for a 500-foot ship) and a width exceeding 4 degrees should be examined very critically.

MANEUVERING

The ability to initiate and check moderate changes in course is one of the most important handling qualities of ships. The 20-20 overshoot maneuver provides an excellent measure of the inherent maneuvering ability of the ship. Two types of criteria for maneuvering are suggested, one for initial turning movement and the other for overshoot. On the basis of the 20-20 overshoot maneuver a 500-foot ship should reach execute heading angle in 65 seconds at 8 knots and 36 seconds at 16 knots. The nomograph in Figure 15 is provided to show criteria for sizes of ships between the range of 300 feet and 700 feet in length and 6 to 20 knots in speed.

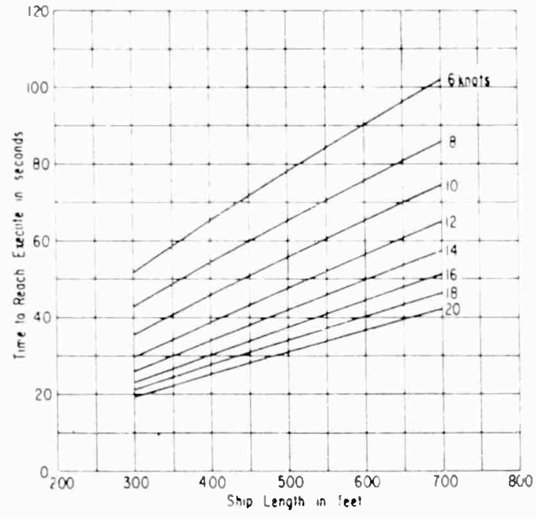


Figure 15 - Nomograph of Criteria for Time to Change Heading in a 20-20 Overshoot Maneuver

The overshoot angle does not vary with size of ship hence a nomograph similar to that for initial turning movement is not necessary. Overshoot angles of 5.5 degrees for 8 knots and 8.5 degrees for 16 knots are suggested as tentative criteria. Straightline interpolations for other speeds may be used as required.

TURNING

It is unrealistic to expect merchant ships to turn as tightly as combatant types of naval ships. Most of such modern naval ships strive for tactical diameter ratios of 3.25 ship-lengths or less with 35 degrees rudder angle. It is believed, therefore, that a tactical diameter ratio of 4.5 ship-lengths is a practicable criterion for merchant types and represents good handling performance objectives. Tactical diameters exceeding 7.0 ship-lengths reflect poor performance qualities and should be tolerated only under special conditions or requirements.

In dimensional terms, the criteria for tactical diameter and advance for various-sized merchant-type ships are shown in Table 3 for speeds of 8 and 16 knots using a rudder angle of 35 degrees.

TABLE 3

Turning Criteria

Ship Length feet	Tactical Diameter yards	Advance yards	Time to Change Heading 180 degrees seconds		Speed Remaining After Changing Heading 180 degrees knots	
			8 knots	16 knots	8 knots	16 knots
300	450	335	207	122	5	10
400	600	450	270	152	5	10
500	750	560	325	185	5	10
600	900	670	377	217	5	10
700	1050	785	428	250	5	10

RECOMMENDATIONS FOR FUTURE STUDIES

The advent of new and improved facilities such as the Rotating Arm and Maneuvering and Seakeeping Basins at the Taylor Model Basin should provide a stimulus for attacking problems in the stability and control of surface ships which have been neglected for centuries. With such facilities and the attendant advances that have been made in instrumentation and test techniques, it should be possible to study handling qualities much more intensively than has been done in this paper. Accordingly, it is recommended that a concerted effort be made to prevail upon the ship owners, shipbuilders, and model basins to accumulate data from definitive maneuvers, particularly on merchant ship types. Such definitive maneuvers should not only be of the type contained herein but should be designed to reveal the handling qualities of ships when subjected to the effects of environment, restricted channels, acceleration and deceleration, and other unusual circumstances.

It is further recommended that, concurrent with the effort to gain a firmer understanding of the status of handling qualities of existing ships, programs should be formulated with the purpose of achieving optimum handling qualities. Such studies can be carried out best in the model basins utilizing research designs where the emphasis will be on optimum stability and control to the exclusion, if necessary, of other characteristics. The advantage of utilizing such an approach is that the work can proceed with an understanding that the ship actually need not be built. It should be possible on this basis to determine what improvements in handling qualities are in store for ships provided that concessions to other requirements do not have to be made. In this manner, the various points of diminishing returns can be defined with reasonable clarity.

Once it is known how good the various handling qualities can be, the designer will be in a much better position to make decisions as to what compromises he is willing to make. It then remains to provide him with the fundamental hydrodynamic data and other design criteria to help him achieve his predetermined end result.

ACKNOWLEDGMENTS

The authors are grateful to Messrs. F. D. Bradley, C. R. Olson, G. W. Williford, Jr., and W. G. Surber, Jr., of the David Taylor Model Basin for performing the model maneuvers within the limited space of the J-Towing Basin. Particular thanks go to Mr. Surber for conducting some of the full-scale trials and compiling the experimental data upon which the numerical studies are based. We are indebted to the Bureau of Ships for permission to use the data herein and especially to Mr. A. Taplin for his keen interest in this work, which led to getting approval for carrying out these special maneuvers on full-scale ships, and for his active participation in some of the full-scale tests.

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ON THE MEASUREMENT OF ADDED MASS AND ADDED MOMENT OF INERTIA OF SHIPS IN STEERING MOTION

by

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Japan

1. DEFINITION OF ADDED MASS AND THE EFFECT OF FREE SURFACE ON THE ADDED MASS

1.1 DEFINITION OF ADDED MASS

It is well known that when a floating body, like a ship, is moving in a fluid, its motion has a character appreciably different from that of the motion in a vacuum, due to the virtual mass effect.

Different definitions are given to this virtual increase of the mass, or added mass; namely,

a) Added mass m_1 as defined from the difference between the moment of inertia in a vacuum $m\dot{v}$ and that in a fluid $(m + m_1)\,\dot{v}$;

b) Added mass m' as defined from the difference between the period of oscillation in a vacuum $2\pi\sqrt{m/k}$ and that in a fluid $2\pi\sqrt{(m + m')/k}$;

c) Added mass m'' as defined from the difference between the momentum in a vacuum mv and that in a fluid $(m + m'')\,v$;

d) Added mass m''' as defined from the difference between the kinetic energy in a vacuum $\frac{1}{2}\,mv^2$ and that in a fluid $\frac{1}{2}\,(m + m''')\,v^2$; etc.

The added masses based upon these different definitions do not always coincide with each other, except in the case of the motion in an ideal fluid without free surface; therefore a value of the added mass by one definition cannot be used as the value of added mass by other definitions.[1] Accordingly, in developing the equation of motion of ships, it is necessary to determine first of all which definition of the added mass or added moment of inertia should be used.

1.2 EFFECT OF FREE SURFACE

Since the free surface has the greatest effect on the added mass, the effect of free surface should be considered first.

It has been disclosed that the effect of free surface on the added mass, when a body in water is accelerated in one direction, is equal to the effect of reverse image of a body to the free surface.[1,2,3] When, there, m_0 is taken as the added mass without free surface and m_1 as the added mass with free surface, then it follows

$$m_1 - m_0 = \text{reverse image effect} = -\text{image effect} \qquad [1]$$

and the virtual inertial resistance equal to $m_1\,\ddot{x}$ is produced in the body.

Let ϕ represent the velocity potential; then the condition of free surface in this case is given by

$$\phi = 0 \quad \text{at} \quad t = 0 \qquad [2]$$

[1]References are listed on page 273.

(This equation holds good as an approximation for any velocity.)

Next, when the steady oscillation is applied to the body in the same direction as it has been accelerated, as mentioned above, the resistance due to wavemaking, and due to bumping with waves which the body has generated, is produced in the body, in addition to the inertial resistance $m\ddot{x}$. When this resistance R is divided into a resistance having the same phase as the velocity of the body and that having the same phase as the displacement of the body, namely,

$$R = K_1 \dot{x} + K_2 x, \qquad [3]$$

then, the equation of motion is given by

$$(m + m_1) \ddot{x} + K_1 \dot{x} + (k + K_2) x = f \cos \omega t \qquad [4]$$

This equation may imply that the virtual increase of restoring force has resulted. Since, however, the body is in the steady motion and thence $\ddot{x} = -\omega^2 x$, the change of the restoring force can be regarded as the change of the virtual inertia. The equation of motion can, therefore, be explained, if the mass is regarded to be

$$m + m_1 - \frac{K_2}{\omega^2} \qquad [5]$$

It follows from the above that the calculation of the period of the body using the equation $T = 2\pi \sqrt{(m_1 + m)/k}$ may not give a result consistent with the experimental data, but, instead, the equation

$$T = 2\pi \sqrt{\frac{m + m_1 - \dfrac{K_2}{\omega^2}}{k}} \qquad [6]$$

should be used.

It should be noted, however, that the inertial resistance when the body is accelerated from the steady state to the nonsteady state is still $(m + m_1)\ddot{x}$ and not $(m + m_1 - K_2/\omega^2)\ddot{x}$.

Next, let us consider the relation between m_1 and m', when putting

$$m + m_1 - \frac{K_2}{\omega^2} = m' \qquad [7]$$

The condition on the free surface of the velocity potential of a body in steady oscillation is

$$\phi = \frac{g}{\omega^2} \frac{\partial \phi}{\partial z} \qquad [8]$$

where ω is the frequency and z is the vertical direction upwards.

It follows that:

1) When the frequency is much greater than the acceleration of gravity, then

$$\phi = 0 \qquad [9]$$

This condition is, therefore, equal to the condition of free surface in case of non-steady acceleration.

Accordingly, the effect of free surface in this case becomes the reverse image effect; that is,

$$\lim_{\omega\to\infty} m' = m_1 = m_0 + \text{reverse image effect} \qquad [10]$$

2) When the frequency is much smaller than the acceleration of gravity, or in other words, in case of slow oscillation, the condition of free surface is

$$\frac{\partial\phi}{\partial z} = 0 \qquad [11]$$

In this case, therefore, the free surface acts as a solid wall, and gives the image effect, namely,

$$\lim_{\omega\to 0} m' = m_2 = m_0 + \text{image effect} = m_1 + 2 \times \text{image effect} \qquad [11\text{-a}]$$

3) For a finite ω between Equations [1] and [2], the effect of free surface becomes very complicated. Experimental results for heaving and pitching have already been given by Gerritsma[6] and by Golovato.[7] Figure 1 gives the results of calculations made by Yamamoto[3] as to submerged cylinder and sphere, which indicates the values of m' for various periods. Figure 1 is shown on the basis of the period made dimensionless. At $T = 0$, m' is smaller than the value without free surface, due to reverse image effect, and with the increase of the period the virtual mass first becomes smaller than m_1, then increases, surpassing m_0, and eventually becomes equal to m_2 as the period becomes infinite.

Figure 2 shows the results of an experiment carried out by the author, in which the additional moment of inertia about the vertical axis of a ship has been obtained from the period of torsional oscillation of a steel bar attached to ship's side. The curve in Figure 2 has a similar tendency to that in Figure 1, and is drawn on the basis of the period made dimensionless, d being the draft.

The record of this experiment clearly indicates that, though J_{z1} varies with the change of the period, only J_{z1} gives the effect on the inertia force for nonsteady acceleration; that is, the following phenomenon is recognized.

In the region of C in Figure 2, or in the case of $J_z' > J_{z1}$, the ship moves for the first half-period of oscillation in the still water without waves generated, as shown in Figure 3 (a); therefore she yaws with the period corresponding to J_{z1}. For subsequent yaws, however, the additional inertia becomes J_z' because of the effect of waves, and the period becomes much slower. On the contrary, in the region of A in Figure 2, the phenomenon is completely reversed, as shown in Figure 3 (b), because $J_z' < J_{z1}$.

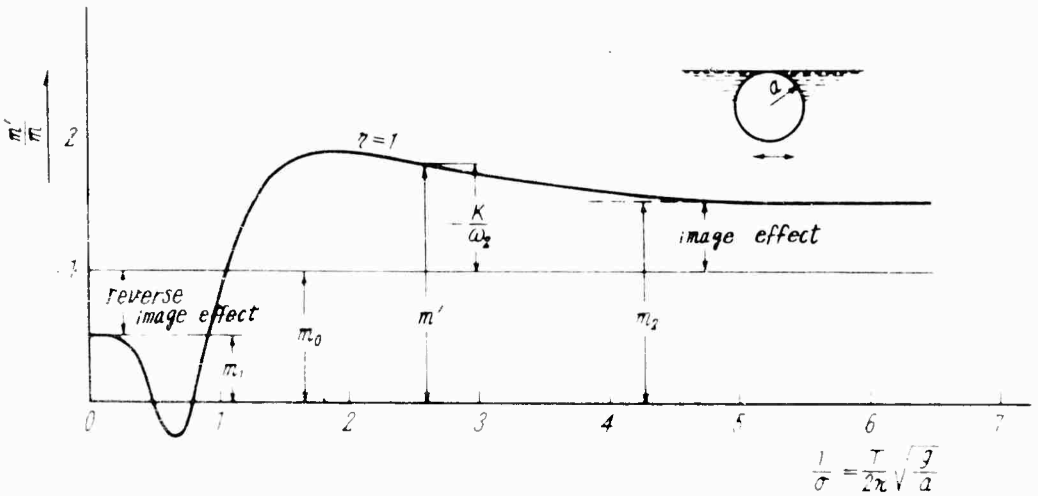

Figure 1 – Added Mass Coefficient of Oscillating Submerged Cylinder

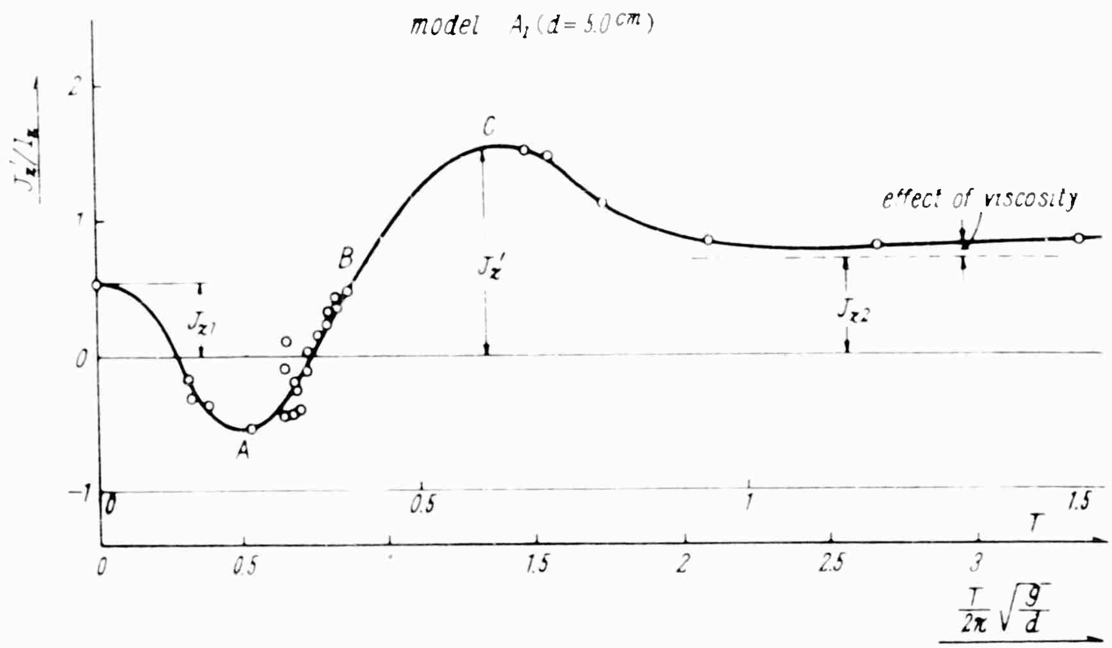

Figure 2 – Measured Values of Added Moment of Inertia for the Yawing

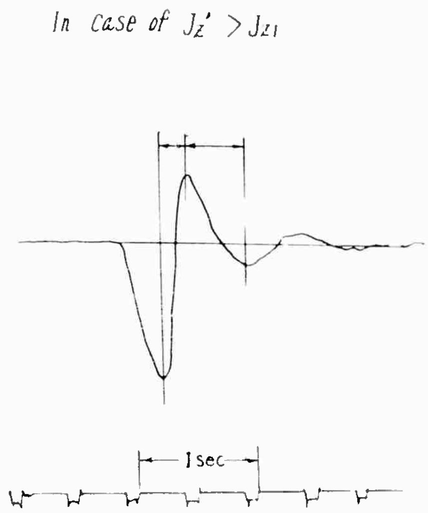

Figure 3 (a)

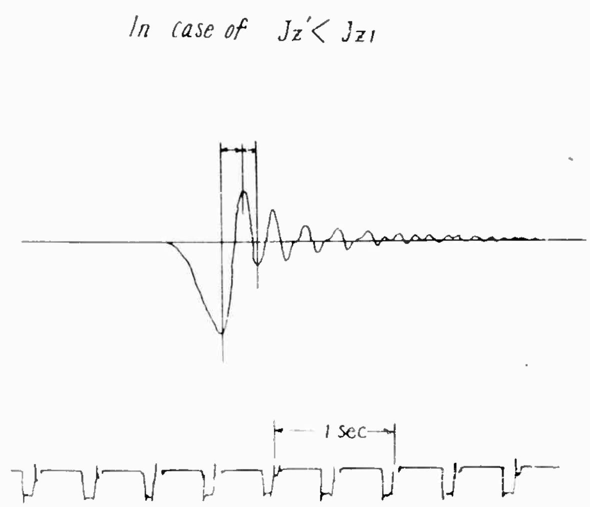

Figure 3 (b)

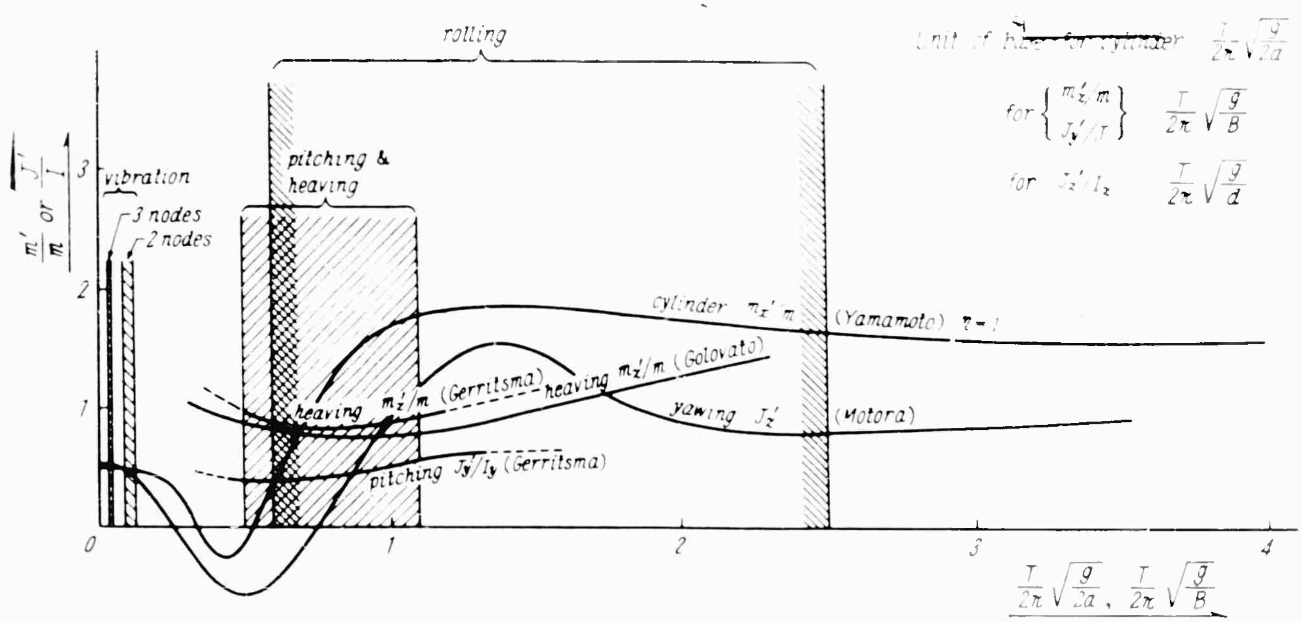

Figure 4 – Change of Added Mass Due to the Change of the Period, and the Range of Periods of Ship Motions

2. THE VALUE OF ADDED MASS TO BE USED FOR THE SHIP'S MOTION IN RESPECTIVE DEGREES OF FREEDOM

Figure 4 indicates the calculated added mass mentioned in the foregoing, the measured added moment of inertia about z-axis given in Figure 2, and the values of added mass obtained by Gerritsma[6] as to heaving and pitching, which are plotted on the basis of the period made dimensionless. These figures show similar tendencies, though there is a little deviation in the positions of the ~~hump and~~ hollow. In Figure 4, the regions of the natural period of rolling of ships, of the natural period of pitching and heaving, and of the period of bending vibration of ships are also shown as shaded portions.

The bending vibration of ships has a very short period both in two and in three nodes, so that the effect of free surface may be considered as the reverse image effect and m_1 and J_1 can be used as the added inertia. The papers,[8, 9, 10, etc.] are dealt with from this viewpoint.

In the case of a rigid body, however, the period covers a wide range, as shown in Figure 4, and therefore a great difference will result according to the adoption of m' and J' or of m_1 and J_1. In the case of rolling with a long period, especially, some added masses are close to J_2. It is therefore necessary to consider the kind of added mass in each degree of freedom which may be most appropriate for expressing the motion.

For this purpose, the ordinate fixed to a ship is taken as shown in Figure 5, and the added mass and moment of inertia about each axis are denoted as given in Table 1, each symbol having the significance as indicated in Figure 1. Strictly speaking, the added mass and moment of inertia which should be included in the term of inertia in the equation of motion for any degree of freedom should be m_1 or J_1, and not m' or J'. However, for rolling or similar oscillations in which the motion in the vicinity of the natural period is evident, it may be more useful to include J_x' in the term of inertia from the beginning. On the contrary, for pitching or heaving or similar oscillations in which damping force is so great that the motion in the vicinity of the natural period is not so evident as that of rolling, it may be advisable to use m' and J' corresponding to the natural period, but, on the other hand, it is considered more reasonable to use m_1 and J_1 from the standpoint that the oscillation due to irregular waves is the response to continual impacts.

As a result of the foregoing considerations, the kinds of added mass to be adopted have been determined, as shown in Table 2.

It should be noted that, in the case of rolling, no serious difficulty will arise if J_x' is substituted for J_{x1}, as I_x is much greater than J_{x1} or J_x', but in the case of pitching or heaving, appreciable difference will result if J_x' is adopted in lieu of J_{x1}.

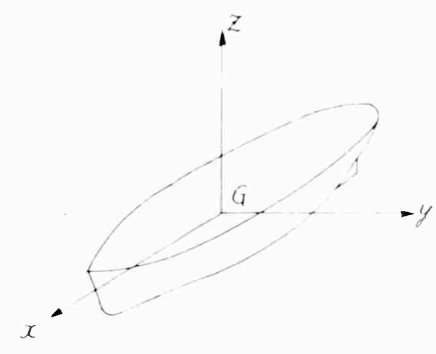

Figure 5

TABLE 1

Kinds of Added Mass and Added Moment of Inertia

	Without Free Surface	As Defined by Inertia	For Arbitrary Period	For Infinite Period
Added Mass				
x-axis	m_{x0}	m_{x1}	m_x'	m_{x2}
y-axis	m_{y0}	m_{y1}	m_y'	m_{y2}
z-axis	m_{z0}	m_{z1}	m_z'	m_{z2}
Added Moment of Inertia				
x-axis	J_{x0}	J_{x1}	J_x'	J_{x2}
y-axis	J_{y0}	J_{y1}	J_y'	J_{y2}
z-axis	J_{z0}	J_{z1}	J_z'	J_{z2}

TABLE 2

		Restoring Force	Damping	Added Inertia / Inertia of Ship	Natural Period	Kind of Added Mass to be Adopted in the Equation of Motion	In Case of Impact and irregular Waves
x-axis	Translation	Nil	Small	Small	Nil	m_{x1}	m_{x1}
	Rotation	Small	Small	Small	yes (long)	J_x'	(J_{x1})
y-axis	Translation	Nil	Great	Great	Nil	m_{y1}	m_{y1}
	Rotation	Great	Great	Great	yes (short)	J_y'	J_{y1}
z-axis	Translation	Great	Great	Great	yes (short)	m_z'	m_{z1}
	Rotation	Nil	Great	Great	Nil	J_{z1}	J_{z1}

3. MEASUREMENT OF ADDED MASS AND ADDED MOMENT OF INERTIA IN STEERING MOTION

The steering motion is aperiodic and is considered as a continual impact. For the equation of motion, therefore, the added mass and added moment of inertia defined by inertia force must be adopted.

3.1 MODEL SHIPS USED

The SS KAMIKAWA MARU (145.00m × 19.50m × 12.20m, d = 8.03m, Δ = 15,825t) was adopted as the parent-form of model ships. From this parent form, model ships with four different Cb's and three different B/D's were made. They are 1.700 meters in length and are made of

wood painted with lacquer. Their particulars are shown in Table 3, and the body plans of the patterns for each group of model ships are illustrated in Figure 6(a), (b), (c), and (d).

TABLE 3

Particulars of Model Ships

Model Ship	A			B			C	D			E
L_m	1.700										
d_m	0.093										
C_b	0.800			0.679			0.565	0.450			0.603
C_p	0.807			0.682			0.599	0.555			0.613
$C_{\otimes}$	0.992			0.983			0.943	0.811			0.983
	A_0	A_I	A_{II}	B_0	B_I	B_{II}	C_0	D_0	D_I	D_{II}	E_0
B_m	0.2280	0.1700	0.3400	0.2280	0.1700	0.3400	0.2280	0.2280	0.1700	0.3400	0.2280
L/B	7.456	10	5	7.456	10	5	7.456	7.456	10	5	7.456
W kg	29.147	21.710	43.420	24.756	18.458	36.916	21.023	16.395	12.224	24.448	22.000
W/L^3	5.933	4.419	8.838	5.039	3.757	7.514	4.279	3.337	2.488	4.976	4.478
I_z kg-m^2	3.32	2.45	4.65	2.65	2.06	3.38	2.30	2.07	1.43	2.35	2.17
k_z/L	0.198	0.197	0.193	0.192	0.196	0.178	0.187	0.209	0.201	0.183	0.184

The relations between C_b, C_p, $C_{\otimes}$ etc. in each group of model ships are determined from the data as given by Yamagata.[11] The model ship E was patterned after Model Ship B_0 but with extremely large cut-up. This model was made to obtain the effect of cut-up by attaching different sizes of deadwood.

3.2 ADDED MASS FOR LONGITUDINAL TRANSLATION m_{x1}

3.2.1 Method of Measurement

Since m_{x1} is considered to be approximately equal to 4 to 8 percent of m, it is necessary to measure the virtual mass $m + m_{x1}$ within the accuracy of 0.2 to 0.4 percent, in order to obtain m_{x1} within the accuracy of ± 5 percent, but such a precise measurement is very difficult.

From the above viewpoint, the following three methods were studied:

1) *Vibration method.* This method is based upon the fact that the limit of the added mass m' obtained from the prolongation of the period of vibration is m_1, when the period becomes infinitesimal, and is intended to obtain the limit of added mass in the infinitesimal period by changing the strength of the spring and thus causing the surging with different periods.

   This method was proved unsatisfactory, however, because it was impossible to produce vibrations with sufficiently short period.

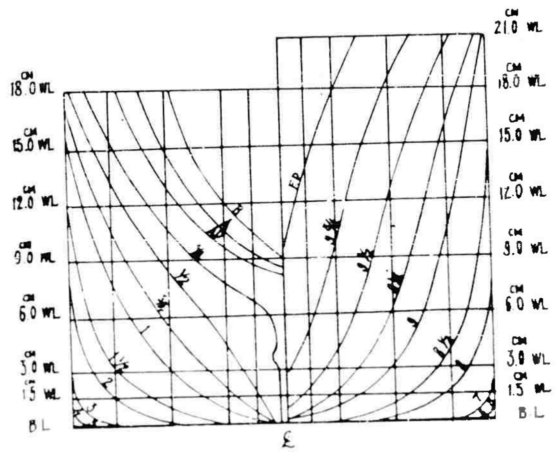

Figure 6 (a) – A_0 Model

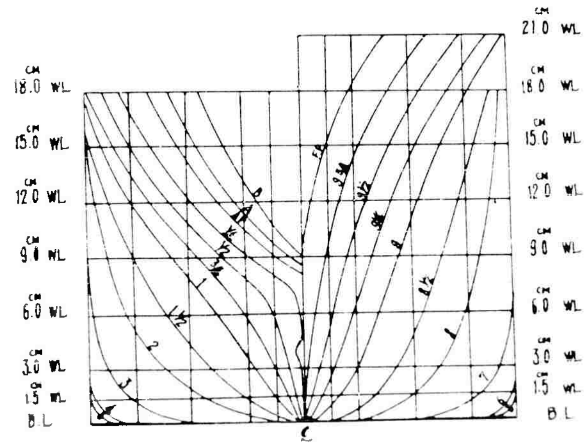

Figure 6 (b) – B_0 Model

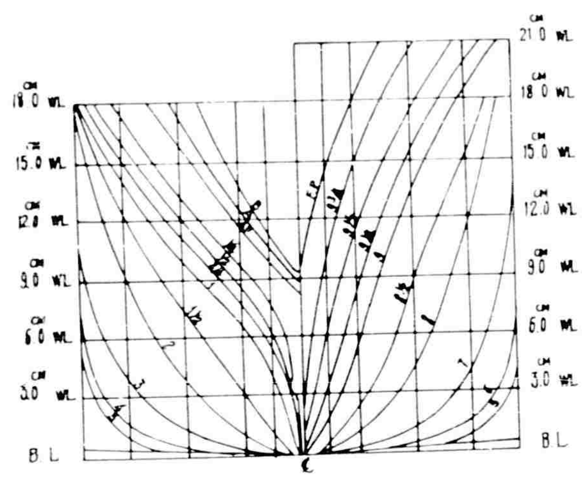

Figure 6 (c) – C_0 Model

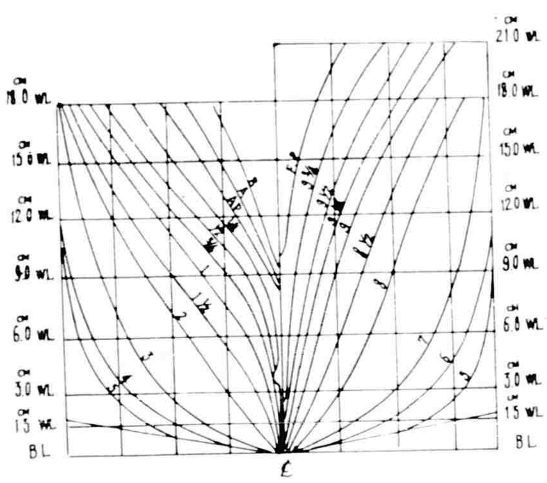

Figure 6 (d) – D_0 Model

2) *Acceleration method.* This method is intended to measure the acceleration when a ship is accelerated by applying a force of known amount, and to obtain the virtual mass from this acceleration.

It was, however, found difficult to measure the acceleration with the required accuracy.

3) *Impact method.* This method is intended to measure the initial velocity caused by the impact of a known amount applied to a ship, and to obtain the virtual mass from this velocity. This method proved satisfactory.

The outline of the impact method is described as follows.

A pendulum was used for giving the impact. As shown in Figure 7, the apparatus is so contrived that the pendulum is made free after having been swung up by an angle α from the upright position, and when it falls to the upright position then it strikes the model ship at the centerline.

Let β = Angle of bouncing of the pendulum,

w_p = Weight of the pendulum,

l_G = Distance from the supporting point to the center of gravity of the pendulum,

K_p = Radius of gyration of the pendulum, and

u_0 = Initial velocity of the ship.

Since the angular velocity of the pendulum changes from $\dot{\alpha} = \sqrt{2gl_G(1 - \cos\alpha)}/K_p$ to $\dot{\beta} = -\sqrt{2gl_G(1-\cos\beta)}/K_p$, the total amount of the impact at a distance from the supporting point is

$$\frac{l_p}{g}(\dot{\alpha}+\dot{\beta})\frac{\cos^2\phi}{l_p} = \frac{w_p}{g}\frac{K_p^2}{l_p}(\dot{\alpha}+\dot{\beta})\cos^2\phi$$

$$= w_p\frac{K_p}{l_p}\sqrt{\frac{2\,l_G}{g}}\left(\sqrt{1-\cos\alpha}+\sqrt{1-\cos\beta}\right)\cos^2\phi \qquad [12]$$

where ϕ is the angle shown in Figure 7.

On the other hand, as the momentum of the ship changes because of this impact by an amount of

$$(m + m_{x1})\,u_0$$

then the virtual mass can be obtained as follows, by putting the above two amounts equal.

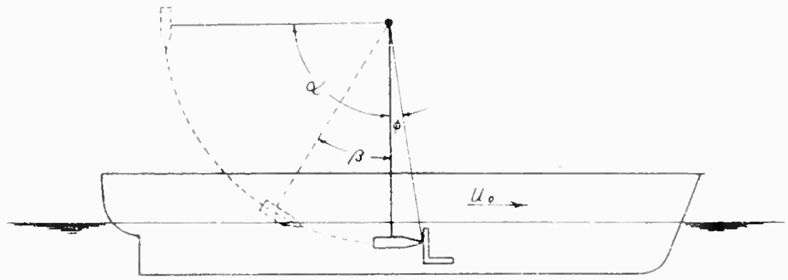

Figure 7

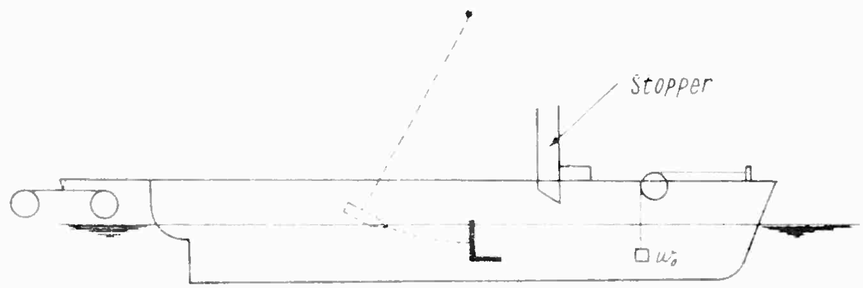

Figure 8

$$m + m_{x1} = \frac{w_p}{u_0} \frac{K_p}{l_p} \sqrt{\frac{2l_G}{g}} \left(\sqrt{1 - \cos\alpha} + \sqrt{1 - \cos\beta} \right) \cos^2\phi \qquad [13]$$

To obtain the initial velocity u_0, a pen was attached to the ship at first to record the movement of the ship on the paper moving in a constant speed, and was attempted to be obtained from tangent at $t = 0$ of the curve thus drawn. However, as the accuracy of this method was not satisfactory, the following method was adopted later.

As shown in Figure 8, a weight w_0 is suspended by a string attached to the stem through a pulley with very small friction, and a stopper is fitted to prevent the ship from moving astern beyond the position of the stopper. When the pendulum is made free after having been swung up by an angle, then the ship begins to move with the initial velocity u_0, due to the impact of the pendulum. The ship is, however, being pulled astern constantly by the weight w_0', so that she returns to the original position after having moved by such a distance h that the initial kinetic energy of the ship is balanced with the increment of potential energy of the weight due to being pulled upwards.

Accordingly, let h represent the distance that the ship has moved or the height that the weight has been raised; then we obtain

$$\frac{1}{2} \left(m + m_{x1} + \frac{w_0}{g} \right) u_0^2 = w_0 h + \delta E \qquad [14]$$

where δE is the work done by the frictional resistance of the ship and the friction of the pulley.

From Equation [13], u_0 is expressed by

$$u_0 = \frac{w_p}{m + m_{x1} + \frac{w_0}{g}} \frac{K_p}{l_p} \sqrt{\frac{2\, l_G}{g}} \left(\sqrt{1 - \cos\alpha} - \sqrt{1 - \cos\beta} \right) \cos^2 \phi \tag{15}$$

From Equations [14] and [15], the virtual mass is obtained as

$$m + m_{x1} = \frac{w_p K_p^2\, l_G}{(w_0 h + \delta E)\, l_p^2\, g} \left(\sqrt{1 - \cos\alpha} + \sqrt{1 - \cos\beta} \right) \cos^4 \phi - \frac{w_0}{g} \tag{16}$$

In this case, the question is the amount of the frictional resistance; if this is very big as compared with w_0, this method cannot be used. As a result of investigations, however, it has been found that the frictional resistance is less than 0.4 percent of w_0 and therefore no difficulty will arise if the amount of frictional resistance is calculated as to each ship and careful correction is made accordingly.

Figure 9 shows an example of the record of the ship's movement, from which h can be measured with sufficient accuracy.

3.2.2 Preliminary Experiment in Air

For the purpose of examining the accuracy of the previously mentioned method, the ship was suspended in the air by two wires, to which an impact was given by the same pendulum as was to be used for the main experiment.

Let l represent the length of the wire, U the impact, and u_0 the initial velocity; then

$$U\, l = m\, u_0\, l$$

$$m = \frac{U}{u_0} \tag{17}$$

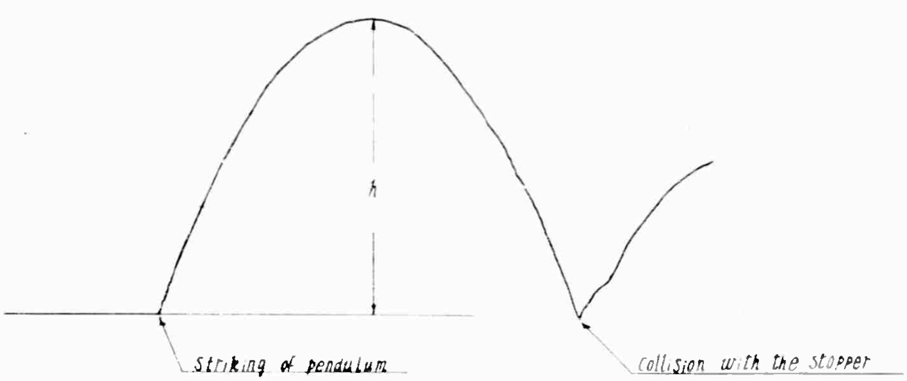

Figure 9

U can be obtained from the angle of bouncing and u_0 from the angular displacement of the ship.

The results of the experiment performed about ten times indicated that the ratio of the measured value to the ship's mass was not exactly 1.000, but 1.031. It was considered that this difference could be attributed to the dissipation of the small portion of the momentum due to the vibration of the ship caused by the impact. Therefore, the results of the experiment in water have been corrected by multiplying this ratio in all cases.

3.2.3 Correction of Error due to Frictional Resistance of Ship

Since the work is done by the frictional resistance, the results of the measurement are in error as if the work done $w_0 h$ by the weight w_0 has been somewhat increased.

Let δE represent this work done and R_{f0} the frictional resistance for initial velocity u_0; then

$$\delta E = \frac{1}{2} R_{f0} h$$

The above equation implies that the error due to the frictional resistance is the same as the error due to the increase of the weight w_0 by an amount $\delta w = \frac{1}{2} R_{f0}$

The calculation of R_f from C_f, using Blasius' equation, indicated that $\delta w / w_0$ was 0.4 percent in the maximum, which cannot be neglected. Therefore, the correction of the error due to the frictional resistance was made by calculating R_f on each ship.

3.2.4 Results Obtained

Measurements were made ten times, first with Model Ship B_0, to examine the scatter of measured values, etc., and then ten times with each type of model ship. These measured values were averaged, excluding the results of the measurement in which the striking of the pendulum was deemed improper.

Figure 10 illustrates these results obtained, in which the values of m_{x1}/m are plotted on the basis of L/B. It may be noted from this figure that m_{x1}/m increases with the increase of L/B. In Figure 10 are also plotted the values calculated by Lamb,[12] as well as those for Inui's theorectical ship's forms S-201 and S-202 calculated by Bessho. There is a little different tendency between the curves for model ships and for prolate spheroid. This difference may be explained by the fact that in the prolate spheriod d/B is constant, whereas in model ships d/B is variable because d was made constant and B was varied.

In Figure 11 m_{x1}/m are plotted on the basis of C_b. This figure indicates that the added mass increases with the increase of C_b.

Figure 12 shows the variation of the added mass for Model Ship B_0, when d/B is varied by changing the draft. From the fact that m_{x1}/m is approximately proportional to d/B, and moreover the values for prolate spheriod at $d/B = 0.5$ and for ellipsoid at $d/B = 0.25$,

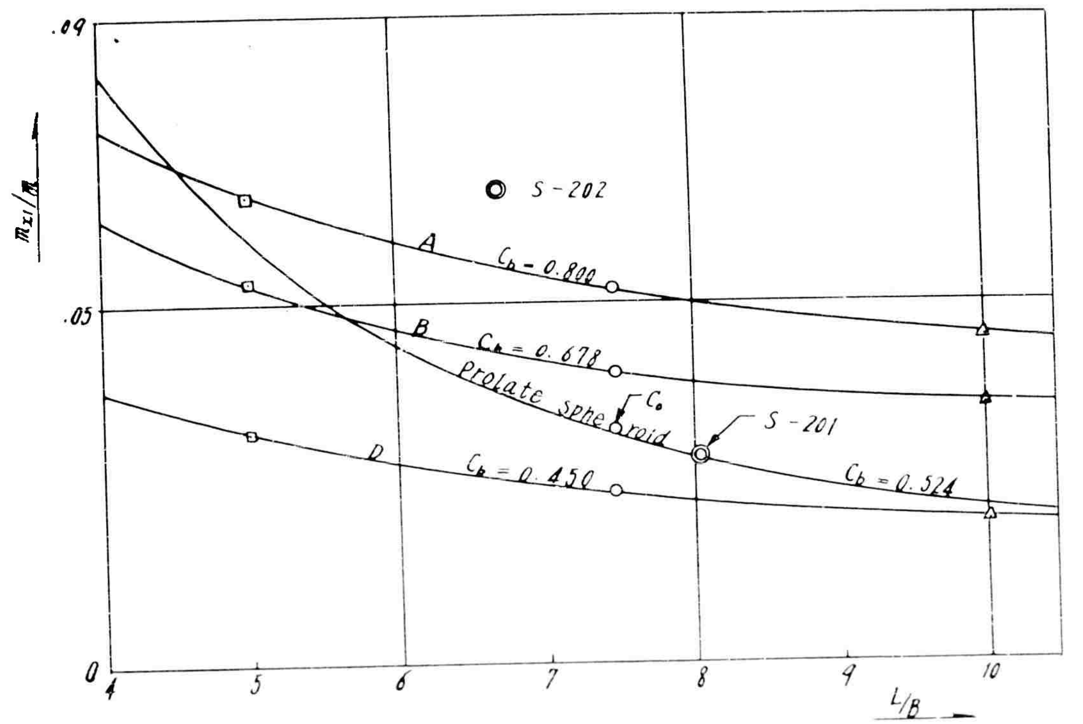

Figure 10

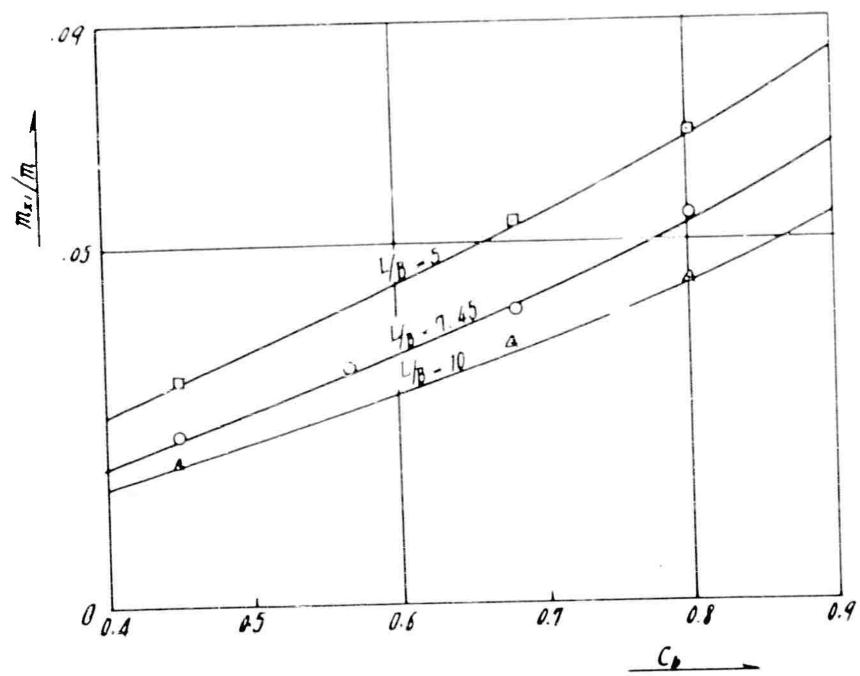

Figure 11

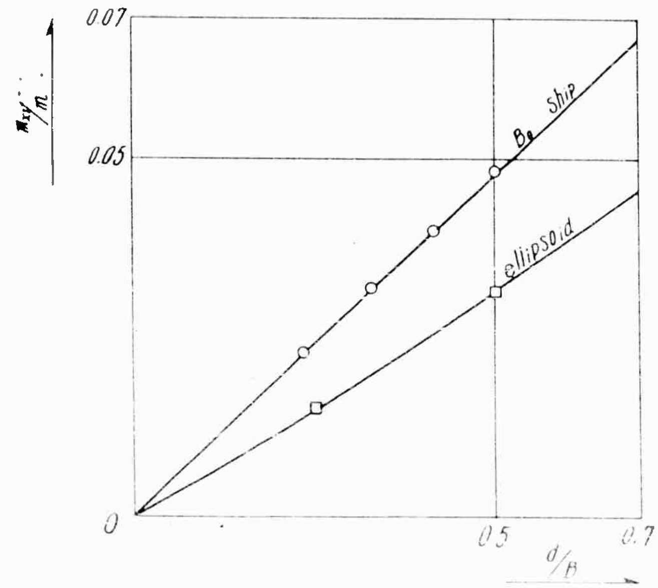

Figure 12

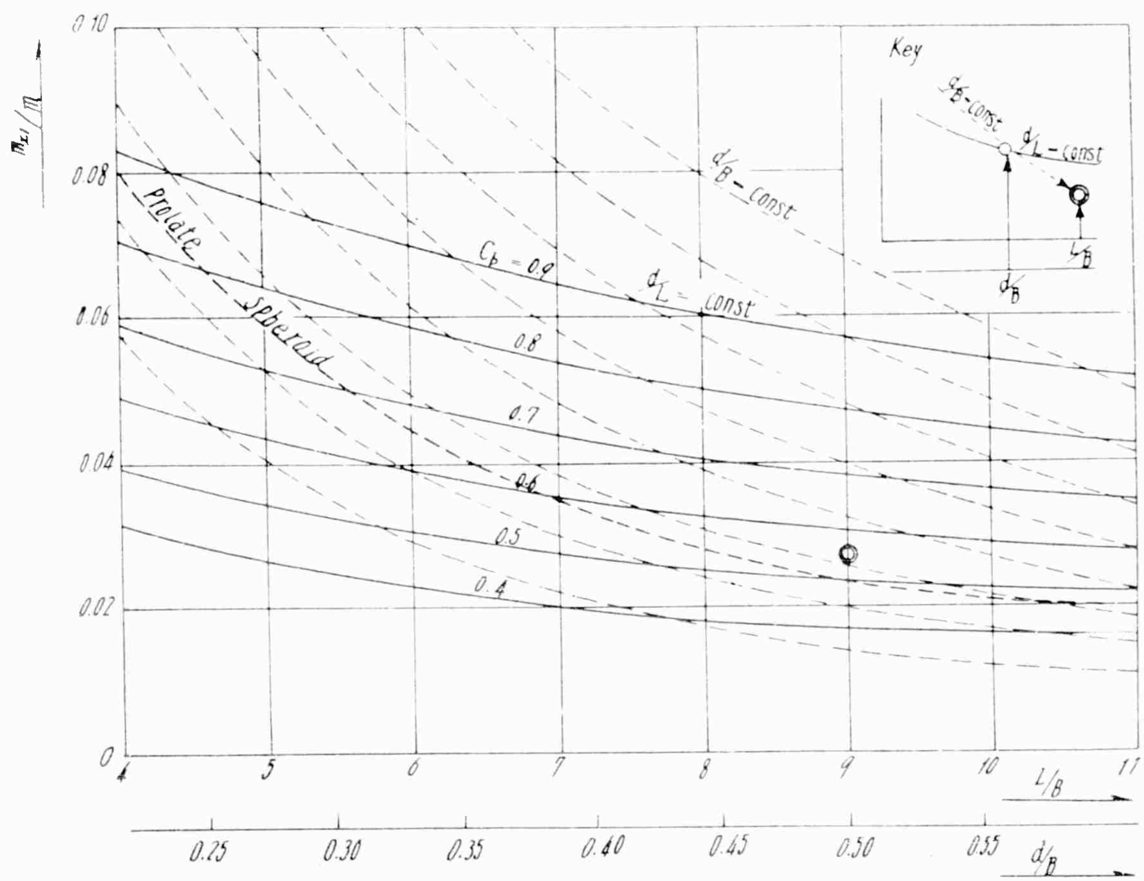

Figure 13 – Longitudinal Inertia Coefficient

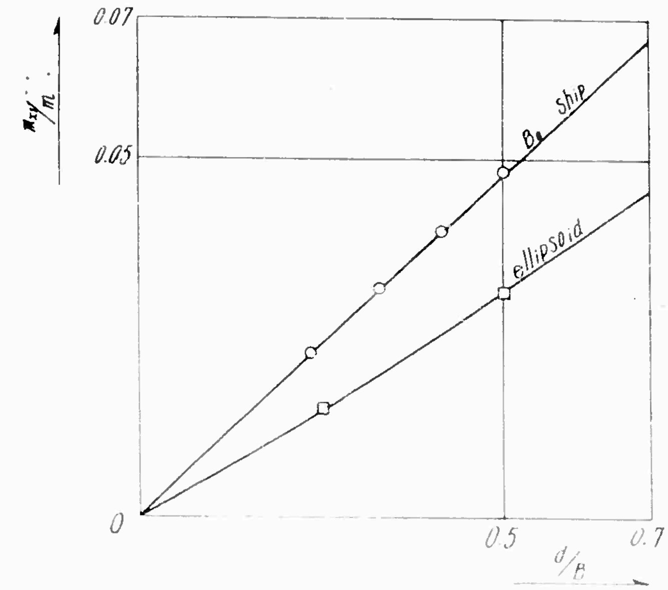

Figure 12

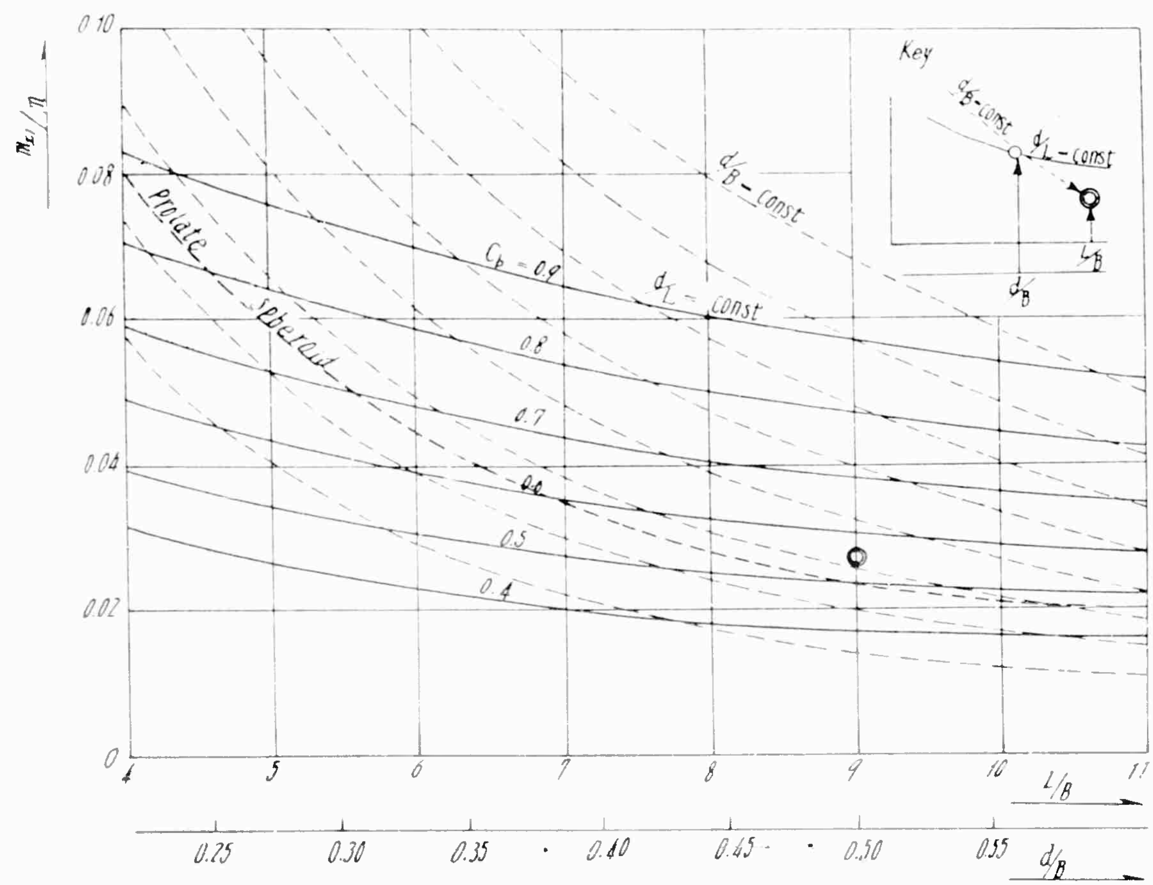

Figure 13 – Longitudinal Inertia Coefficient

plotted in the same figure are also approximately proportional to d/B, it is considered that m_{x1}/m is approximately proportional to d/B.

Figure 13 indicates the variation of m_{x1}/m for constant d/B obtained from the above relation. When the values of m_{x1}/m for each L/B at the points of $C_b = 0.4, 0.5 \ldots\ldots\ldots 0.9$, which are obtained from the readings of Figure 11, are plotted on the basis of L/B, a group of solid lines in Figure 13 for constant d can be drawn having the parameter of C_b.

The broken lines in Figure 13 are a group of curves for constant d/B (= 0.75), which have been drawn using the relation given in Figure 12 (m_{x1} is proportional to d/B). From these curves, m_{x1}/m for arbitrary L/B and d/B can be obtained in the following way.

Take d/B corresponding to the desired draft at the base, read the value of m_{x1}/m at this d/B on the solid line corresponding to given C_b, draw a curve through this point parallel to the nearest broken line (with constant d/B) and read the value of m_{x1}/m on this curve at given L/B; then the value thus obtained indicates m_{x1}/m for given L/B, C_b, and d/B.

The thick broken line in Figure 13 is for the prolate spheroid ($d/B = 0.5$). Since this curve is approximately parallel to the curves with constant d/B, and moreover the values of m_{x1}/m are approximately equal to the values for $C_b = 0.5$ (C_b of the prolate spheroid is 0.5236), the theoretical values of m_{x1} for the prolate spheroid are considered to be applicable to ships with similar range of C_b, though on ships with greater C_b the values are appreciably different.

3.3 ADDED MASS FOR SWAYS m_{y1}

3.3.1 Method of Measurement

To examine the suitable method of measurement, the vibration method, acceleration method, and impact method were studied in the same way as the added mass m_{x1} for longitudinal translation has been measured. Because of a great damping, however, only the acceleration method was found fairly satisfactory; the other methods were not practicable.

Therefore, a different method was considered, based upon the phenomenon that when a slender body is accelerated in the water the direction of force and acceleration do not coincide with each other.

In Figure 14, when a force F is applied to a ship at her center of gravity (or precisely, virtual center of gravity) from a direction having an angle α fro the centerline of the ship, then the acceleration to x-direction is

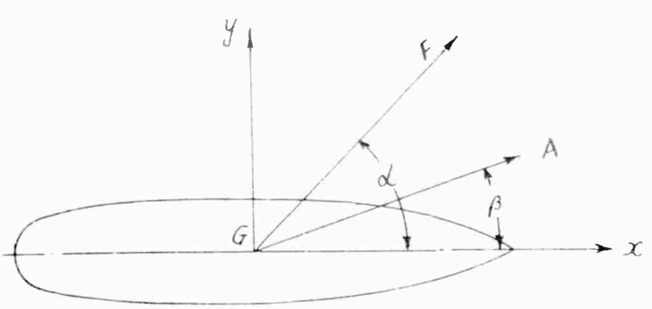

Figure 14

$$\ddot{x} = \frac{F \cos \alpha}{m + m_{x1}} \qquad [18]$$

and the acceleration to y-direction is

$$\ddot{y} = \frac{F \sin \alpha}{m + m_{y1}} \qquad [19]$$

Therefore, when β is taken as the angle from the centerline of the ship to the acceleration A, then

$$\tan \beta = \frac{\ddot{y}}{\ddot{x}} = \frac{m + m_{x1}}{m + m_{y1}} \tan \alpha \qquad [20]$$

This equation indicates that the direction of the acceleration becomes closer to the centerline of the ship than does the direction of the force. When $\tan \alpha$ and $\tan \beta$ are measured, the ratio $m + m_{y1}$ to $m + m_{x1}$ can be calculated from Equation [20], and if m_{x1} is known m_{y1} can be obtained.

In order to obtain m_{y1}, using the above relation, an apparatus as shown in Figure 15 was made. This apparatus is so contrived that the ship is fitted with strings at her centerline on the deck and the bottom, which are stretched in the parallelogram shape to prevent the ship from rolling, and then connected to a stopper on one side and pulled by a weight through a pulley on the other side. A midget light is attached to the ship directly above her center of gravity. The ship and strings are painted white so that they can be taken in the photograph clearly. The photograph was taken from a position directly above the ship when she was at rest, and then the movement of the light when the string was disconnected from the stopper was also taken in the same film by means of double exposure.

The results obtained are as examplified in Figure 16 (a), and (b), from which the ratio $\tan \alpha$ to $\tan \beta$ can be read with sufficient accuracy. Table 4 shows the results of an experiment which was carried out to see whether Equation [20] actually holds good, and in which the direction of the force was varied in a wide range for this purpose. Figures 16 (a) and (b) are the examples of the records for α = 16.4 deg and α = 57.6 deg, respectively.

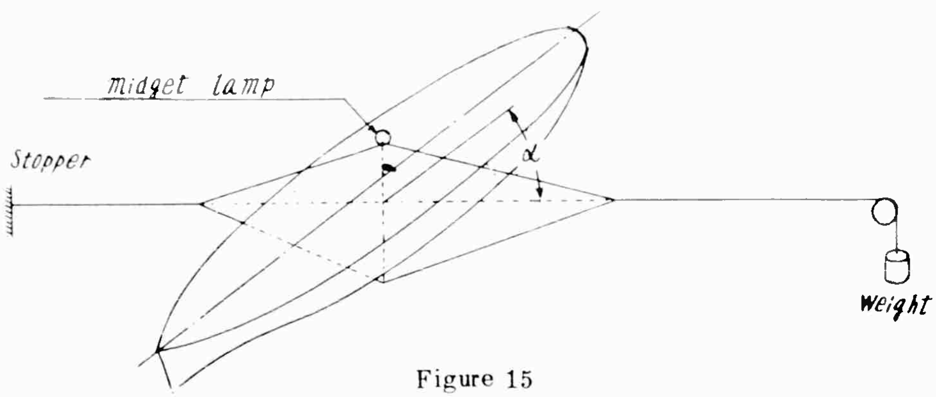

Figure 15

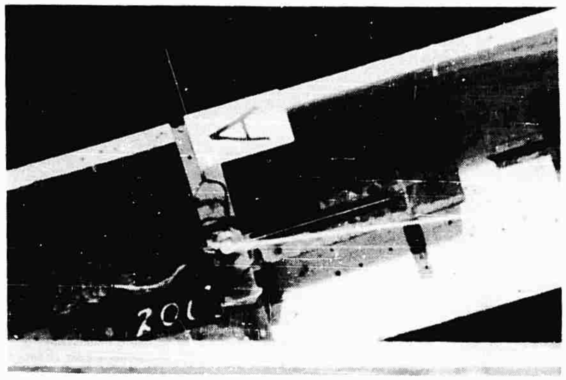

Figure 16a

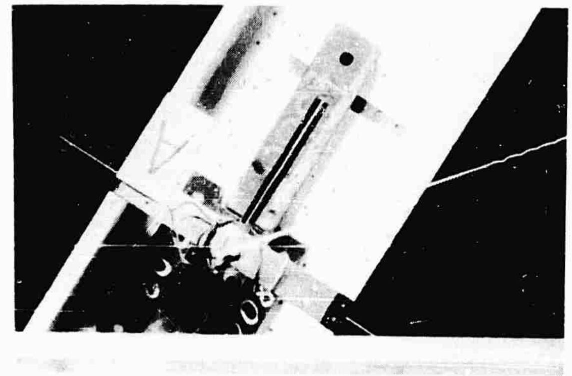

Figure 16b

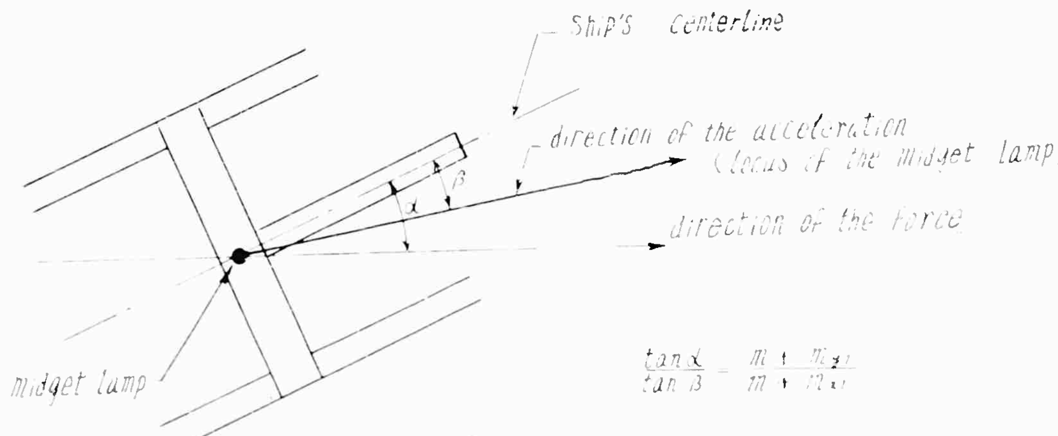

Figure 16c

Figure 16

TABLE 4 – Model Ship B_0

α	$\frac{\tan\alpha}{\tan\beta} = \frac{m + m_{y1}}{m + m_{x1}}$	$\frac{m + m_{x1}}{m}$	$\frac{m + m_{y1}}{m}$	$\frac{m_{y1}}{m}$
16.4	1.880			
22.2	1.890			
36.2	1.885			
43.5	1.89			
57.6	1.888			
82.5	1.910			
average	1.891	1.040	1.967	0.967

Since, in the actual measurement, it was difficult to measure tan α tan β when α is very large or very small, experiments were made several times in such a manner that α might become approximately 35 deg, and the results were averaged.

In order to determine whether the results thus obtainod agree with those by more direct method, experiments by acceleration method were also made. The apparatus of this method is as shown in Figure 17. The ship is fitted with a guide having small friction to prevent the ship from rolling, and strings attached to the ship's sides are stretched in athwartship direction, and connected to the stopper on one side and pulled by a weight w through a pulley on the other side. The difference of acceleration at the instant that the stopper is disconnected is recorded by the oscillograph.

Let $\ddot{y}_0$ represent the maximum acceleration; then m_{y1} can be obtained from

$$\left(m + m_{y1} + \frac{w}{g} + \delta m\right) \ddot{y}_0 = w \qquad [21]$$

where δm is the increment of added mass due to the inertia of the pulley.

An example of the oscillogram is shown in Figure 18. High accuracy cannot be expected from this result, because the vibration of the guide, etc., has been superimposed in the curve of acceleration recorded. The results have a scatter of about $\pm$ 5 percent, though the average agrees fairly well with the results by the foregoing method, and therefore the foregoing method is considered more reliable from the viewpoint of accuracy.

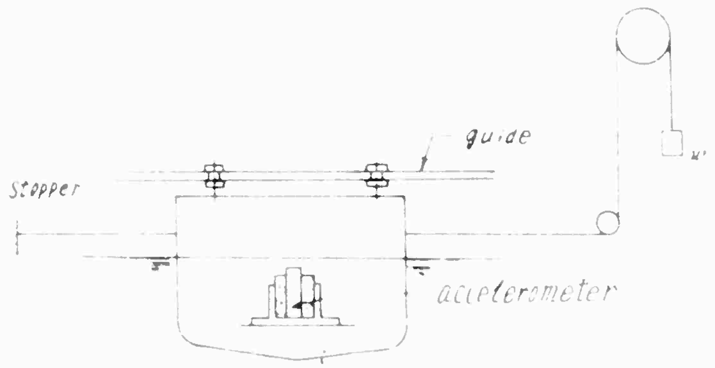

Figure – 17

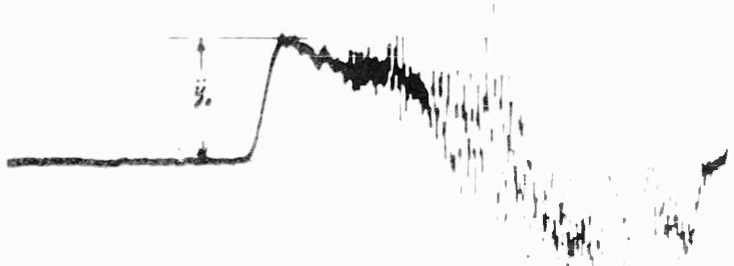

Figure – 18

3.3.2 Results Obtained

1) Measured values

Results of the experiments carried out several times with Model Ship B_0 to examine the accuracy of the measurement were that the accuracy was approximately $\pm$ 2 percent.

The results of the measurement made with each model ship are as shown in Table 5, in which only averages are given.

TABLE 5

Model	$\tan\alpha / \tan\beta$	$(m + m_{x1})/m$	$(m + m_{y1})/m$	m_{y1}/m
A_0	1.782	1.0503	1.873	0.873
A_I	2.092	1.0448	2.186	1.186
A_{II}	1.524	1.0643	1.622	0.622
B_0	1.896	1.0400	1.972	0.972
B_I	2.250	1.0356	2.330	1.330
B_{II}	1.610	1.0526	1.695	0.695
C_0	1.975	1.0324	2.404	1.040
D_0	2.092	1.0234	2.141	1.141
D_I	2.543	1.0315	2.623	1.623
D_{II}	1.757	1.0297	1.810	0.810

2) Relation with L/B

In Figure 19, the values of m_{y1}/m shown in Table 5 are plotted on the basis of L/B. It may be noted from this figure that, as L/B increases or the ship becomes slender, m_{y1}/m increases radically. The broken line in Figure 19 represents the calculated value for the prolate spheroid without free surface. A different tendency is seen between this curve and the curves from model ships. The difference may be attributed to the fact that the prolate spheroid has a constant d/B and variable L/B, whereas in the model ships the measurements were made with constant d/L. This difference will be referred to later.

3) Relation with C_b

Figure 20 shows the relation with C_b. It may be noted from this figure that m_{y1}/m decreases with the increase of C_b, but this may be due to the decrease of the sharp portions of the ship.

4) Relation with d/B

Since Figure 19 shows the results of experiments with constant d and therefore having variable d/B with the variation of L/B, it does not give the effect of the change in L/B or d/B only. Accordingly measurements were made with drafts varied as to model ship A and B to examine the effect of d/B only.

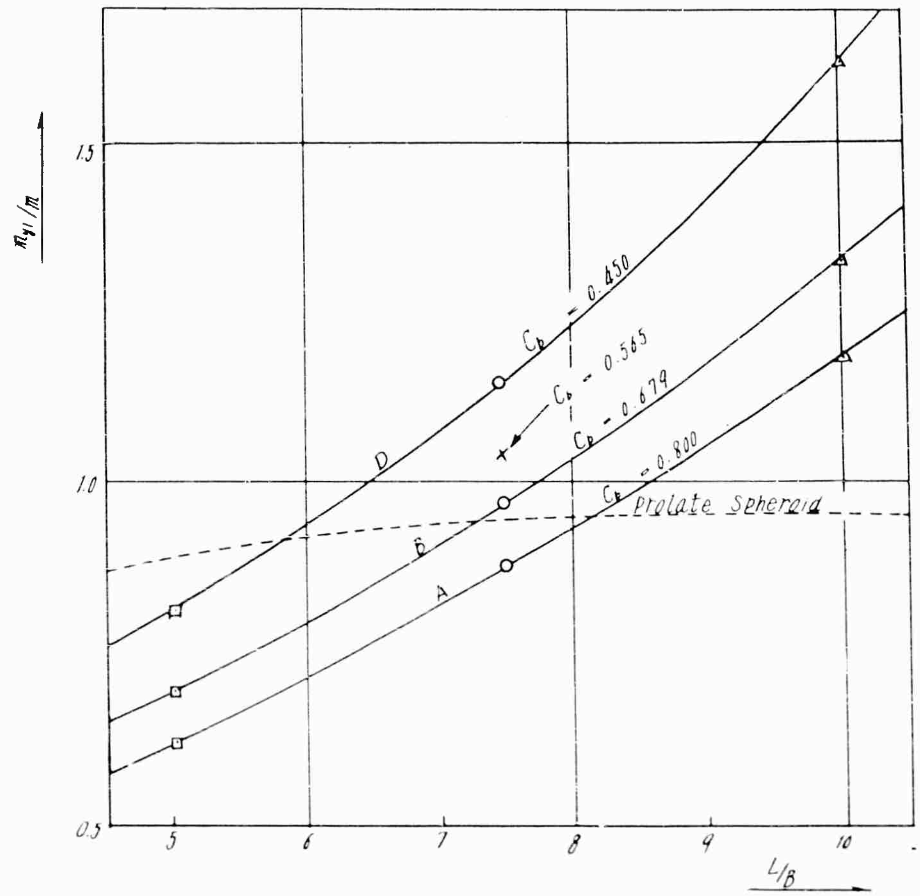

Figure 19

The results are as shown in Figure 21, which indicates that, contrary to the results for m_{x1}, m_{y1}/m is not exactly proportional to d/B, but that it increases with a regular tendency as d/B increases. Accordingly, d constant curves in Figure 19 can be modified approximately to d/B = constant curves using the relation as given in Figure 21.

Figure 22 shows the results of this modification, in which d/L constant curves have been drawn by plotting m_{y1}/m read off from curves corresponding to $C_b = 0.4 - 0.9$ in Figure 20. The broken lines in Figure 22 are d/L = constant curves, which have been drawn so as to intersect the solid lines at $L/B = 7.45$ (standard draft) and by modifying, from the relation in Figure 21, the deviation due to the change of d/B.

From Figure 22 the values of m_{y1}/m for arbitrary L/B, C_b, and d/B can be obtained in the following way. Take the desired d/B of the ship on the base, read the value at this d/B on the d/L = constant curve corresponding to give C_b, draw a curve from this point parallel to the nearest d/B = constant curve, read the value on this curve at given L/B, then the value thus obtained indicates m_{y1}/m for given L/B, C_b and d/B.

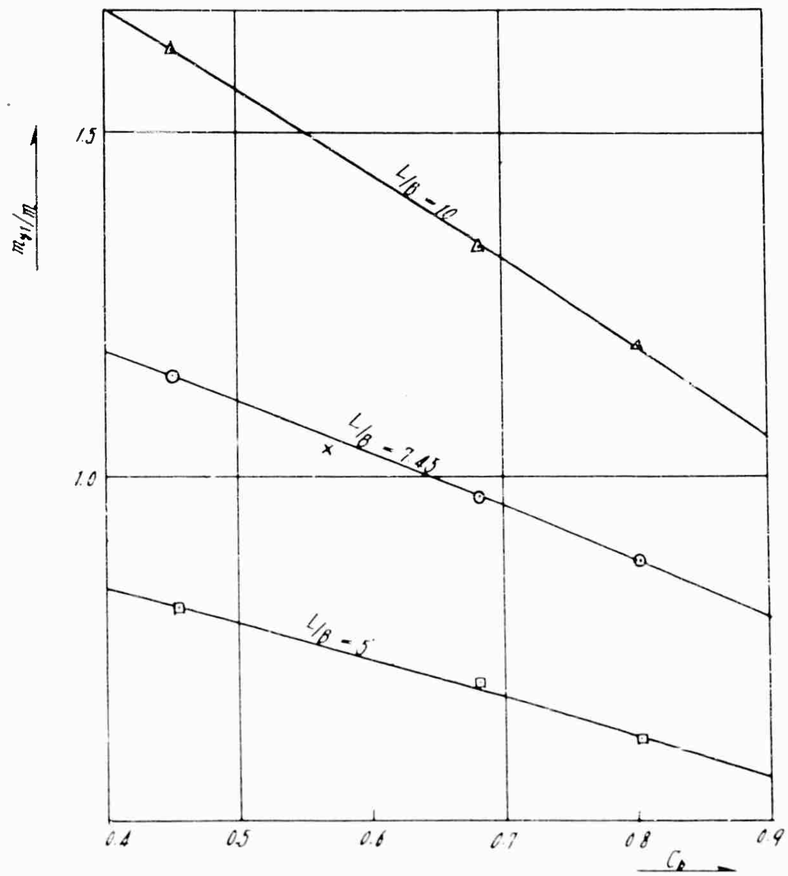

Figure 20

5) Comparison with theoretical values for prolate spheroid

The calculated values of m_{y1}/m for the prolate spheroid with d/B = constant = 0.5 are plotted in Figure 22, which, contrary to the case of m_{x1}/m, have not the similar tendency as that of model ships, even when the results of experiment have been modified to correspond to d/B = constant. In case of ships particularly, m_{y1}/m tends to increase as L/B increases, even when d/B is constant. It is considered that this difference is due to the increase in the sharpness of bow and stern as L/B increases. When d/B = constant curve, which corresponds to the curve for the prolate spheroid with C_b = 0.514 and d/B = 0.5, as drawn from Figure 22, a chain line a little below the uppermost d/B = constant curve is obtained, which shows that m_{y1}/m is much greater than that of the prolate spheroid. This difference is considered to be attributed also to the sharp form at the bow and stern. Accordingly, the theoretical value for the prolate spheroid cannot apply to ships as far as m_{y1} is concerned.

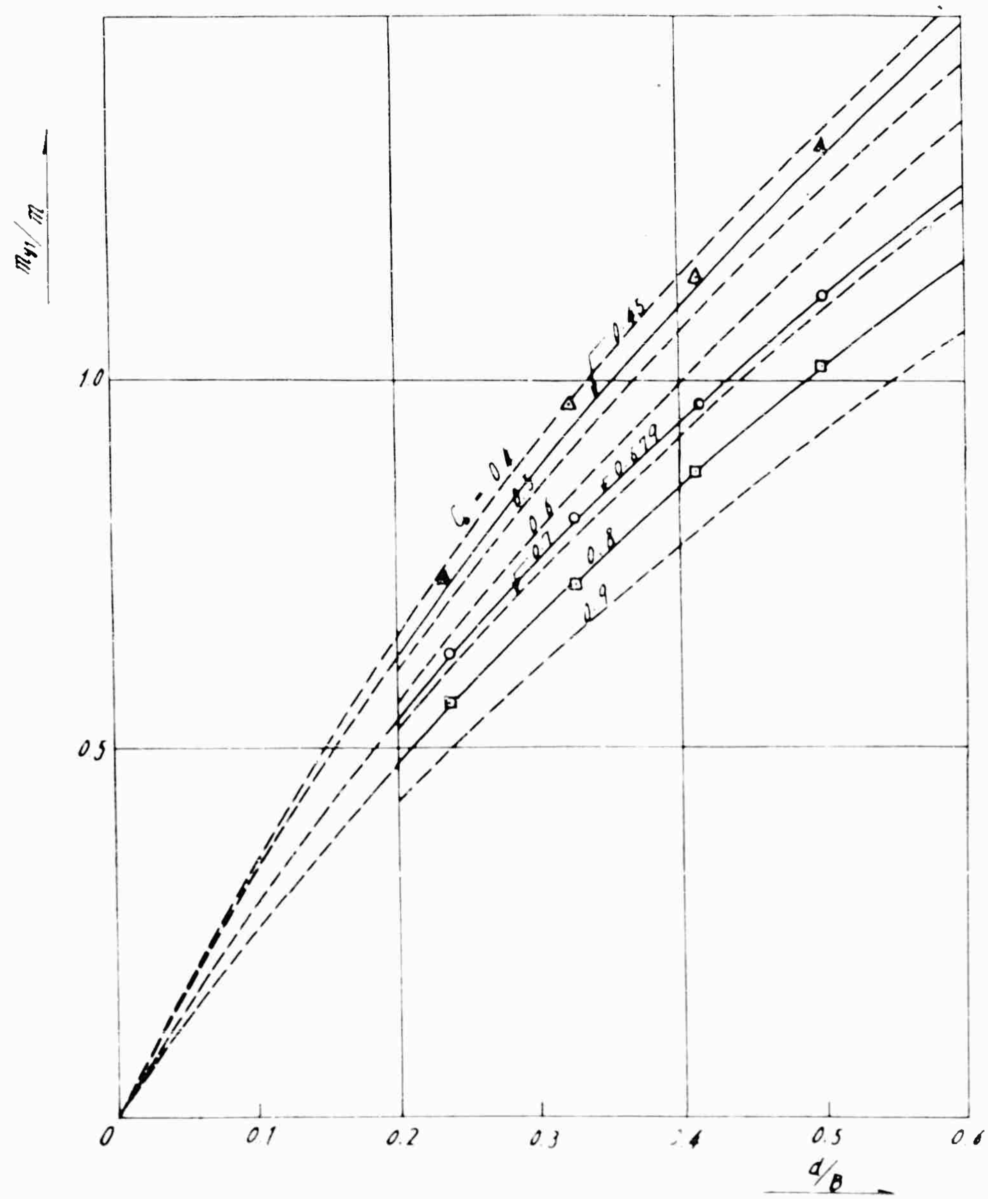

Figure 21

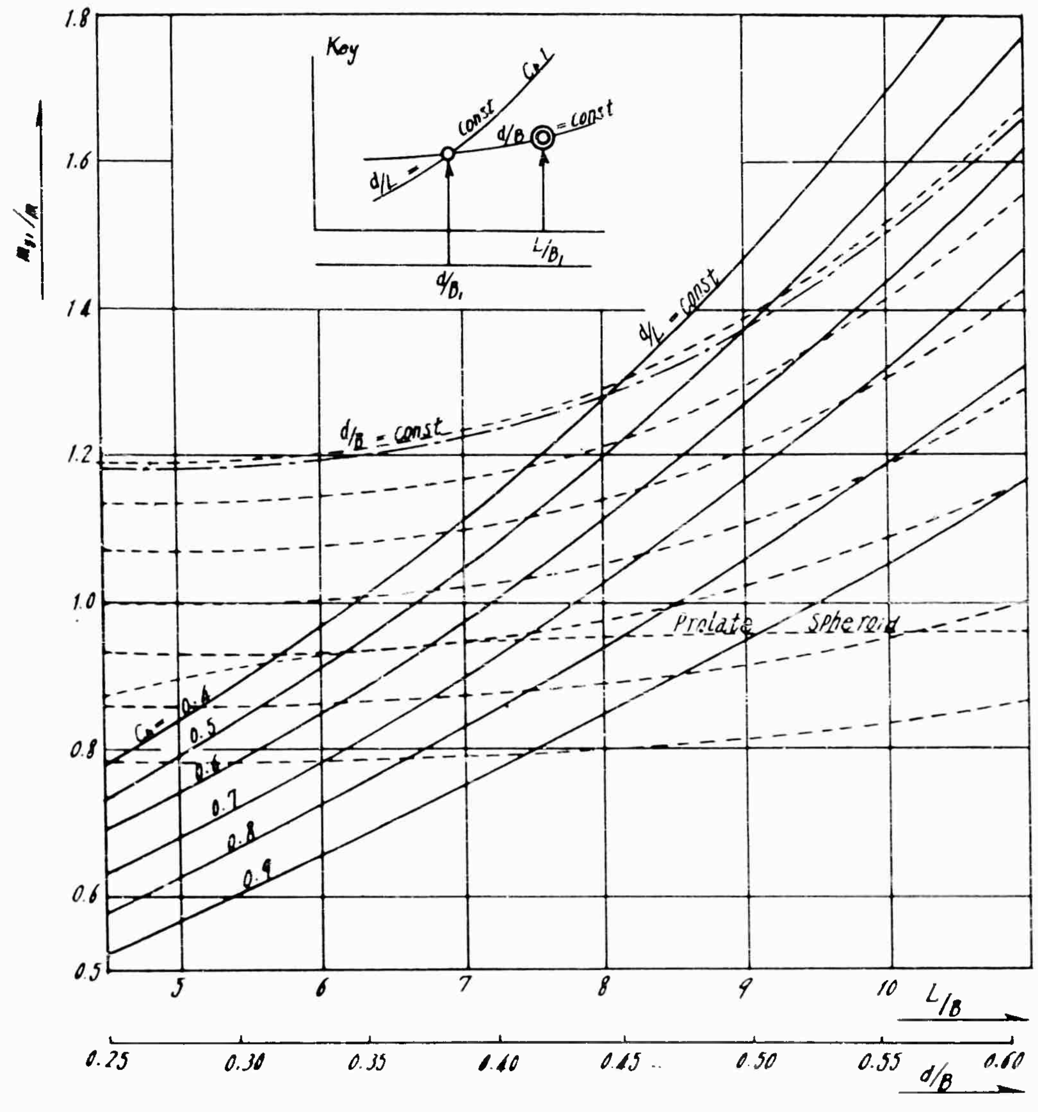

Figure 22 – Transverse Inertia Coefficient

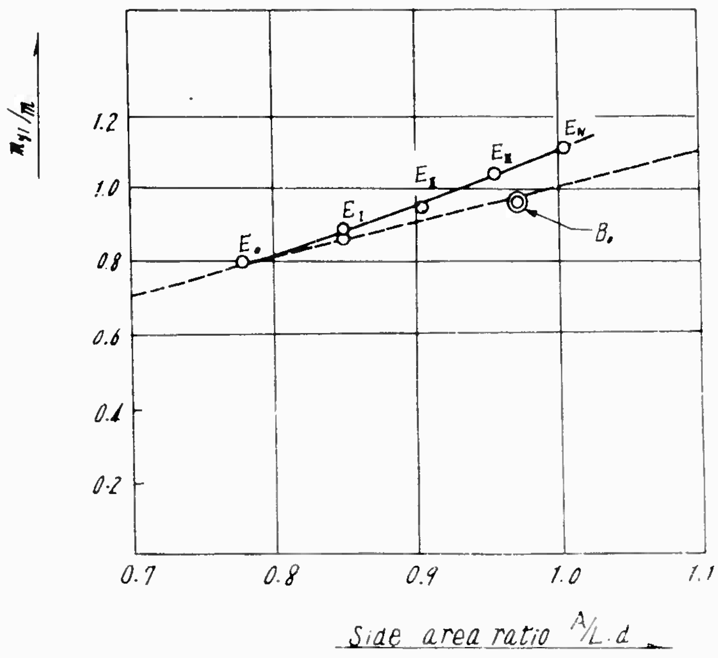

Figure 23

6) Relation between lateral area and m_{y1}

In order to examine the change of m_{y1}/m when lateral area is varied by changing the area of deadwood, experiments were made with Model Ship E_0 fitted with four kinds of fins. The results are as shown in Figure 23.

Fins are fitted symmetrically to the bow and stern. The broken line in Figure 23 has been drawn on the assumption that m_{y1}/m is proportional to the lateral area. The measured values are somewhat greater than these calculated values. This difference can be anticipated as a matter of course, because of the fact that the deadwoods are made from thin plank. In the approximate estimation, however, the increase of m_{y1}/m due to the increase of lateral area may be considered to be proportional to the lateral area; namely,

$$\frac{\delta m_{y1}}{\delta A} = \frac{m_{y1}}{A} \qquad [22]$$

where δm_{y1} is the increment of m_{y1} and δA is the increment of lateral area.

3.4 ADDED MOMENT OF INERTIA FOR ROTATION ABOUT Z-AXIS J_z

3.4.1 Method of Measurement

Similar to the case of 3.2, the vibration, acceleration, and impact methods were examined, and as a result the impact method was adopted. As shown in Figure 24, the apparatus is so contrived that the pendulum is made free, after having been swung up by an angle α, and

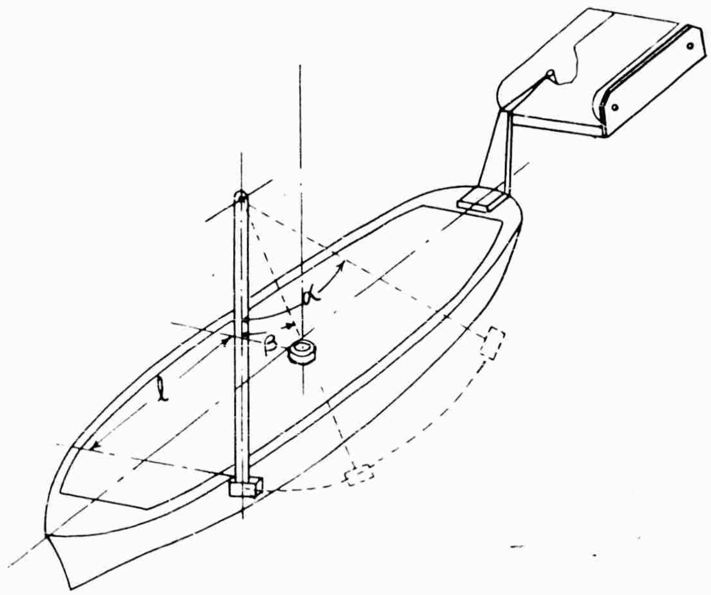

Figure 24

when it falls to the upright position, then it strikes the ship at a position having the distance l from her center of gravity.

Let

β = Angle of bouncing of the pendulum,

w_p = Weight of the pendulum,

l_G = Distance from the supporting point to the center of gravity of the pendulum,

K_p = Radius of gyration of the pendulum,

$\dot{\theta}_0$ = Initial velocity of the ship,

W = Weight of the ship,

I_z = Moment of inertia of the ship,

l = Distance from the center of gravity to a position at which the pendulum strikes.

Since the angular velocity of the pendulum changes from $\dot{\alpha} = \sqrt{2gl_G(1-\cos\alpha)}/K_p$ to $\dot{\beta} = -\sqrt{2gl_G(1-\cos\beta)}/K_p$, the total amount of the impact at a distance from the supporting point is

$$\frac{I_p}{g}(\dot{\alpha}-\dot{\beta})\frac{l}{l_p} = \frac{w_p}{g}K_p^2 = \frac{l}{l_p}(\dot{\alpha}-\dot{\beta}) \qquad [23]$$

On the other hand, as the momentum of the ship changed due to this impact by an amount of

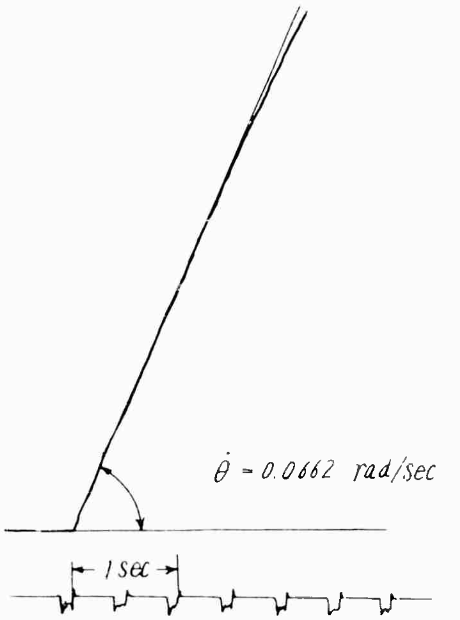

Figure 25 (a)

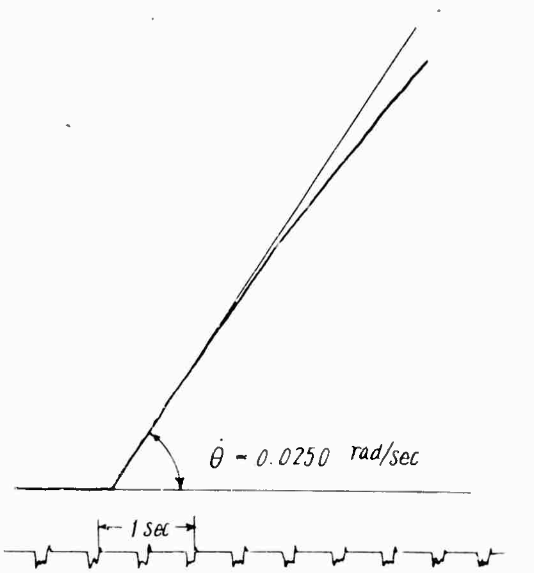

Figure 25 (b)

$$\dot{\theta}_0 (I_z + J_z) / g$$

then $I_z + J_z$ can be obtained as follows by putting the above two amounts equal.

$$I_z + J_z = w_p \; K_p^2 \; l \, (\dot{\alpha} - \dot{\beta}) \, / \, l_p \, \dot{\theta}_0 \qquad [24]$$

α and β can be obtained from the photograph with long exposure, and can be recorded by a pen attached to the stern of the ship.

Figure 25 shows as example of θ_0 recorded. The damping of the velocity, which had been a matter of concern before the experiment, has been found not so great as to obstruct the measurement of the initial velocity, and therefore θ_0 could be obtained with sufficient accuracy.

In order to examine the accuracy of this method, comparisons were made between I_z obtained by the impact in the air with various swing angles of the pendulum and I_z obtained by the vibration method. The results are as shown in Table 6.

TABLE 6

l cm	40				
α deg	90.0	90.0	90.2	61.2	61.6
β deg	21.6	23.2	22.7	9.3	10.8
$\dot{\theta}_0$ rad/sec	0.0576	0.0578	0.0581	0.0355	0.0354
$I_z \cdot g$ kg-m^2	2.79	2.83	2.82	2.84	2.85
$I_z \cdot g = 2.770$ kg-m^2 by oscillation method					

The reason why I_z by impact method is always somewhat greater than I_z by vibration method may be that some portions of the momentum are dispersed due to the vibration, etc. The accuracy of this method is, however, satisfactory for the purpose of the present investigation, so that this method has been adopted.

3.4.2 Results Obtained

The results obtained are shown in Figures 26 –30. Figure 26 indicates the change of J_{z1} due to the change of C_b, expressed in the form of J_{z1}/I_z. Since, however, I_z changes according to the weight distribution, it may not be appropriate to compare the added moment of inertia in the term of J_{z1}/I_z. From this viewpoint, Figure 27 has been drawn, which indicates the change of added moment of inertia expressed in the form of added radius of gyration K_{z1} ($= \sqrt{J_{z1}\, g/W}$) over the length of the ship L. It may be noted from this figure that the added moment of inertia increases as the ship becomes fine.

J_{z1} is greater than has been anticipated, which may be partly attributed to the increase of acceleration due to the deadwood fitted at the bow and stern. Figure 28 gives the change of K_{z1}/L due to the change of L/B, which indicates that the added moment of inertia increases with the increase of L/B Since J_{z1} at the extremity of $L/B = \infty$ must correspond to J_{z1} for a plane of the shape equal to the center plane of the ship, the curve for any type of model ship tends to converge to a finite value. On the other hand, J_{z1} must become 0 at $L/B = 1$. In fact, the curve for any type of model ship converges to 0 at $L/B = 1$.

Figure 29 gives the change of the virtual radius of gyration when the draft is varied, which indicates that the added moment of inertia decreases intensely as the draft decreases.

In the present investigation, the effect of the virtual moment of inertia due to the change of L/d has not been dealt with, but the general tendency can be presumed from the curve in Figure 29 for the same type of model ship and with different drafts.

Figure 30 shows a chart derived from the foregoing results, from which the values of K_{z1}/L for arbitrary L/B, d/L, and C_b can be obtained in the same manner as explained in Figure 13 and Figure 22.

Figure 31 indicates the change of the added mass due to the change of cut-up, which has been obtained from the experiment with Model Ship E_0 fitted with four kinds of deadwood. From this figure the tendency towards increasing the added mass due to fitting of the deadwood can be presumed.

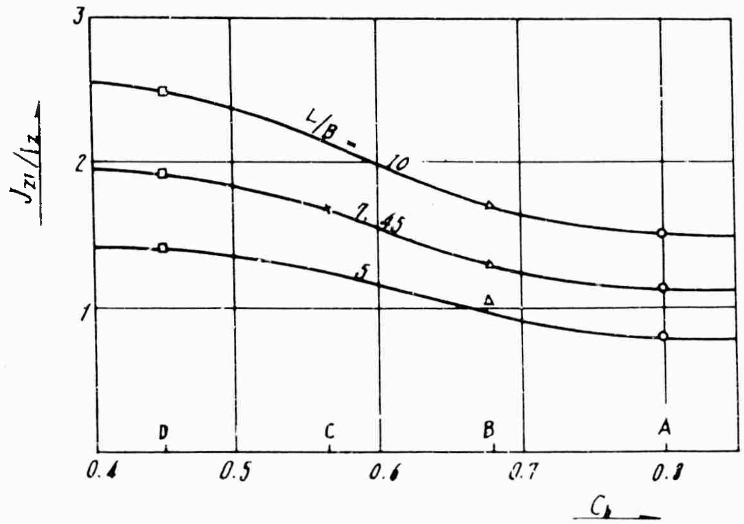

Figure 26 – The Ratio of Added Moment of Inertia to Ship's Moment of Inertia

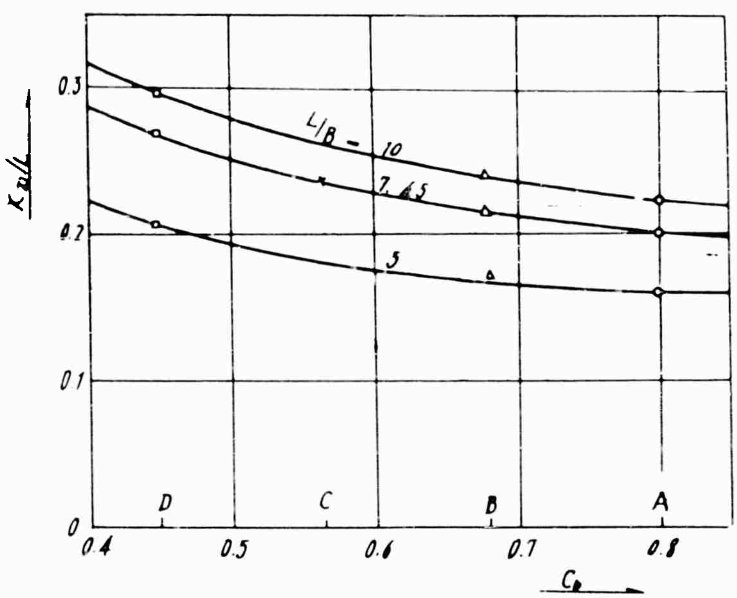

Figure 27 – Change of Added Radius of Gyration with the Change of C_b

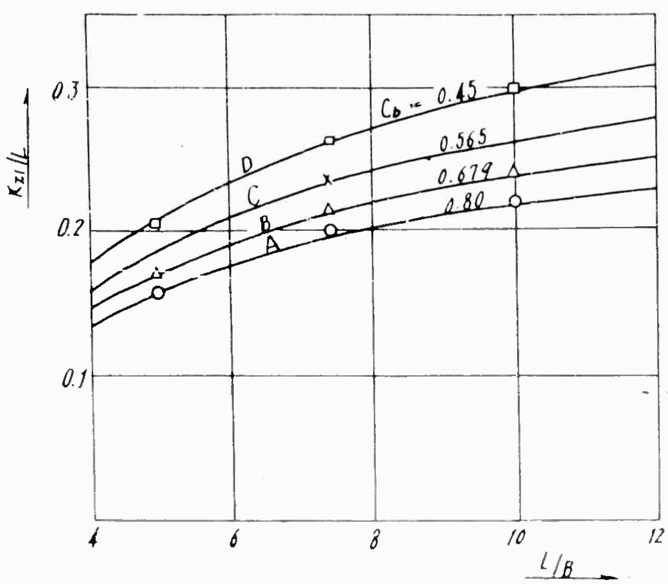

Figure 28

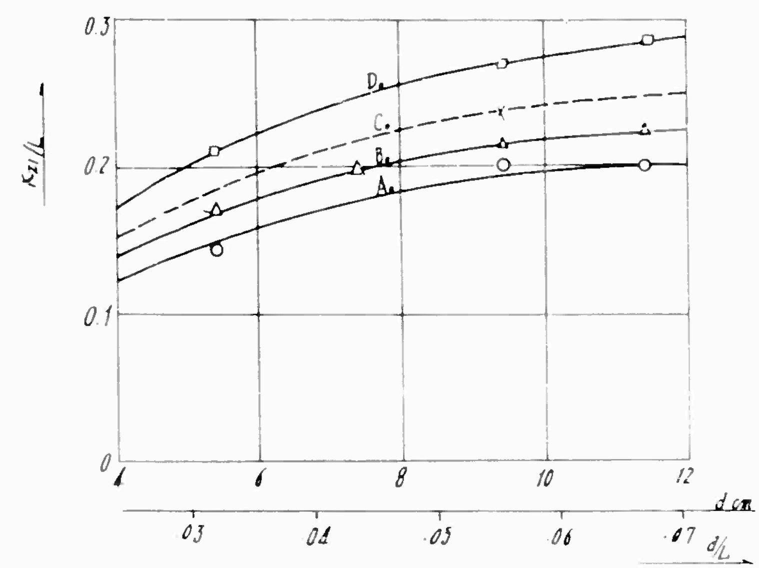

Figure 29

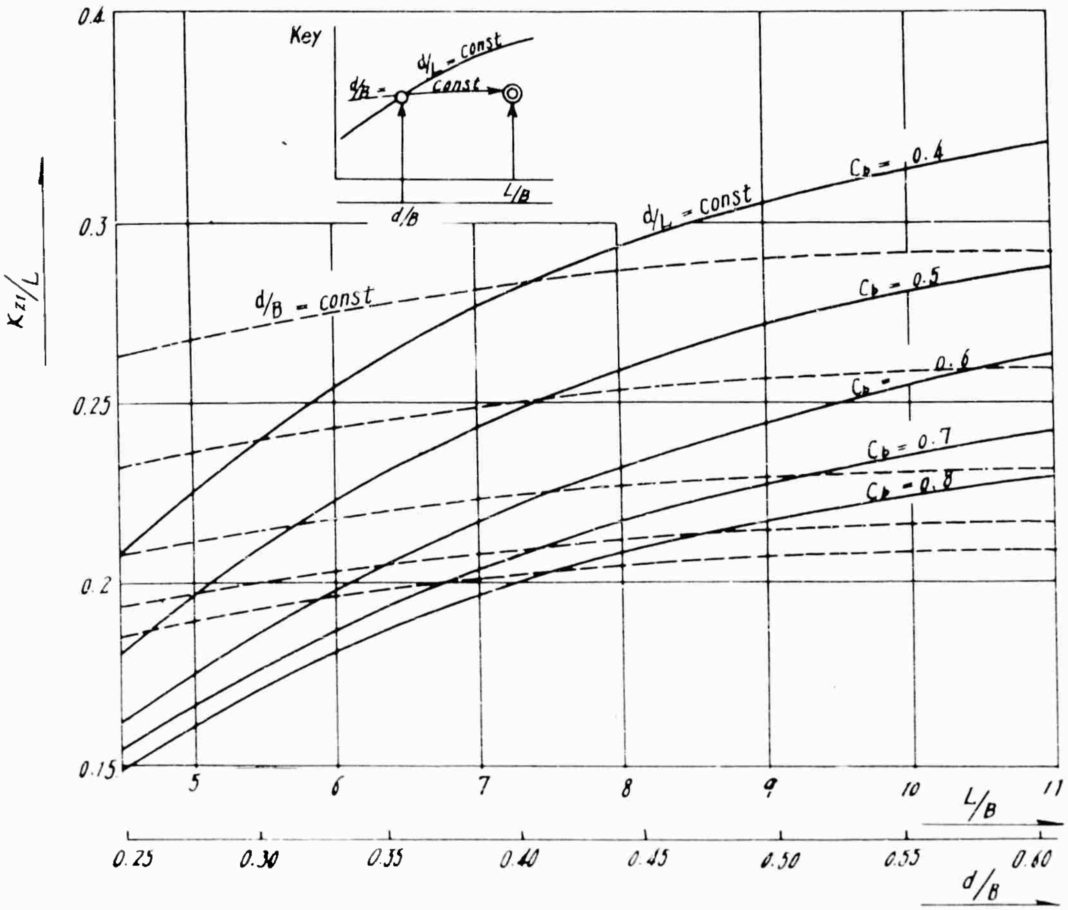

Figure 30 – Added Radius of Gyration about Z Axis in Fraction of Ship Length

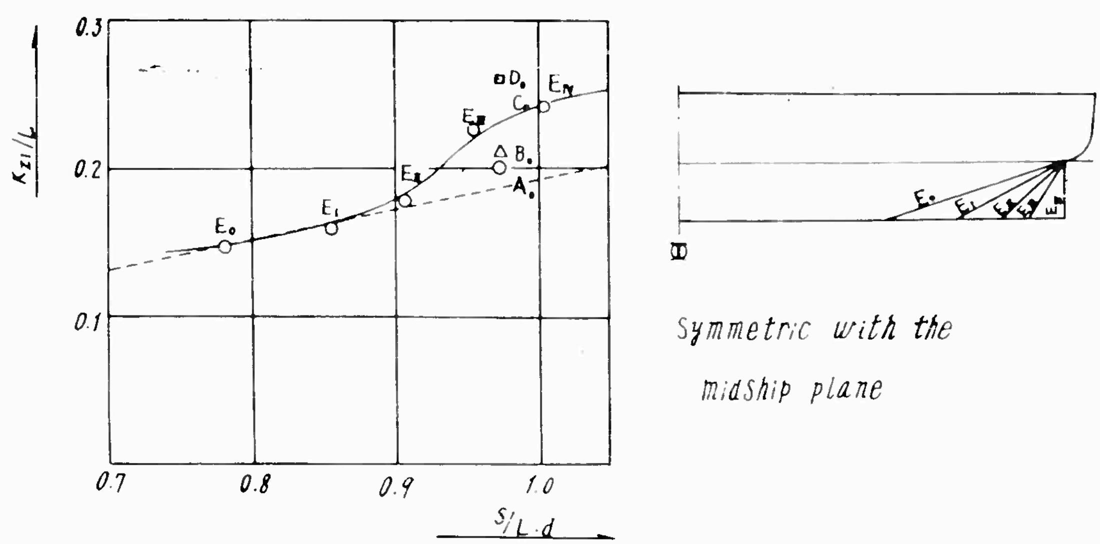

Figure 31

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ANALYSIS OF KEMPF'S STANDARD MANEUVER TEST AND PROPOSED STEERING QUALITY INDICES

by

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ABSTRACT

In previous papers, the author has shown that steering motion of a ship may be practically described by a first-order equation of motion

$$T \frac{d\dot{\theta}}{dt} + \dot{\theta} = K\delta$$

where $\dot{\theta}$ and δ denote turning angular velocity and helm angle, respectively. The index K is a ratio between a steady turning angular velocity and a corresponding helm angle, and represents turning ability of a ship. The other index T represents stability on course and quick-responsiveness in steering, and relates closely to Davidson's stability index p_1. Performance of a ship in steering may be determined well by these two indices.

An analysis of Kempf's standard maneuver test employing the equation determines the indices K and T of a ship concerned through a simple calculation, and then yields a general representation of her steering quality.

The present paper provides a new practical procedure for the analysis and also deals with a proposed formulation of steering quality indices K and T as functions of hull forms and relative rudder sizes, which has been obtained from the analyses for about 70 actual ships and some free-self-propelled models with several alternative rudder sizes.

INTRODUCTION

For a long time an important problem has been to determine what measure of describing the maneuverability of a ship is reasonable and how to obtain it. Although turning trials have been carried out for large numbers of ships, the propriety of constructing the measure of maneuverability merely from a ship's performance in steady turning with hard-over helm is doubtful. The actual process of maneuvering is generally not such a steady turning but rather a succession of transient phases of turning maneuvers with helm angles put to starboard and to port at random.

Pointing out these circumstances, Kempf proposed another maneuver test procedure named as "standard maneuver test" or "serpentine test," and gave also two kinds of figure of merit on maneuverability which relate respectively to a ships' travel for finishing the standard maneuver and to an overswinging angle of heading after a rudder is put to the opposite side.[1]

The author attempted to interpret Kempf's maneuver test employing the first-order simulating equation of motion, developed by the author,[2] so as to obtain from the test a new measure of maneuverability which is not merely a relative measure, like Kempf's but which describes

[1]References are listed on page 304.

the general character of a ship in steering quantitatively. The proposed steering quality indices K and T together constitute such a measure of maneuverability; K represents turning ability and T represents stability on course and quick response in steering. These abilities are just the fundamental elements of maneuverability. Provided with the indices K and T, the motions of a ship under a given arbitrary form of steering may be predicted fairly accurately.

Section 1 of this paper discusses maneuverability in terms of the indices. Section 2 provides a procedure for determining the indices from Kempf's maneuver test through an easy calculation, and also illustrates the results of the analyses for approximately seventy actual ships These indices are naturally functions of hull form, relative rudder size, and other factors. Section 3 relates to a systematic formulation of the indices as functions of these particulars on the basis of the actual data, and then provides a procedure of estimating roughly the indices for a given ship. Section 3 also includes several important considerations on the constitution of steering motion.

1. FIRST-ORDER EQUATION OF MOTION IN STEERING AND A REPRESENTATION OF MANEUVERABILITY USING STEERING QUALITY INDICES K AND T

1.1 FIRST-ORDER EQUATION OF MOTION IN STEERING

Recent studies on ship steering motion which employ equations of motion have shown their validity and usefulness.[3,4] These equations are of the form of simultaneous linear differential equations relating to drifting motion coupled with turning angular motion of a ship in steering. The drift angle is, however, so small, relatively, that steering motion of a ship may be described substantially only by defining heading angle of a ship as a function of time. Then, eliminating the drift angle from the simultaneous equations of motion, we obtain a single equation of motion of the following form:

$$T_1 T_2 \frac{d^2\dot{\theta}}{dt^2} + (T_1 + T_2)\frac{d\dot{\theta}}{dt} + \dot{\theta} = K\delta + KT_3 \frac{d\delta}{dt} \qquad [1]$$

where $\dot{\theta}$ is the turning angular velocity of a ship;

δ is the helm angle as a function of time; and

K, T_1, T_2 and T_3 are the coefficients* composed of the coefficients of the original equations and depending on hull form, relative rudder size, and other factors of a ship.

This equation describes the steering motion of a ship just as do the original equations of motion so far as turning angular motion is concerned.

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*More detailed discussions are shown in Reference 2.

Although the equation can be widely utilized in research on the steering problem, more concise description is desirable, particularly for practical purposes. Examining the equation from such a point of view, it is found that the equation may be simulated by the first-order equation as follows:

$$T \frac{d\dot{\theta}}{dt} + \dot{\theta} = K\delta \qquad [2]$$

where $T = T_1 + T_2 - T_3$. This is the first-order equation of motion in steering. It is discussed in more detail in Reference 2.

1.2 INTERPRETATION OF KEMPF'S STANDARD MANEUVER RESULTS EMPLOYING THE FIRST-ORDER EQUATION IN STEERING

Kempf's standard maneuver is practiced in the following way, as is well-known:

a. Set a certain angle of helm (e.g., 15 deg) to starboard;

b. When ship's course deviation has reached this angle to starboard, reverse the helm to the same angle to port;

c. When the course deviation has reached the same angle to port, reverse the helm again to the same to starboard, and so on.

While actual maneuvers of ships vary too much to be represented by a single form of steering, Kempf's standard maneuver may be typical of normal maneuvers. Consequently a reasonable approach may be to examine the reliability of the first-order Equation of Motion [2] referring to the standard maneuver results for various types of ships. Figures 1–7 illustrate several typical results of these analyses. Chain lines in these figures represent calculated ship motions, using the equation of motion with those K and T values which are determined under the hypothesis that an observed ship motion may be described by an equation such as Equation [2]. The practical procedure of determining these K and T values will be discussed later in detail. If this calculated ship motion simulates well the observed ship motion, we can confirm the hypothesis and then the reliability of the first-order Equation of Motion [2] and we also may consider these K and T values as those of a particular ship.

The present interpretation of steering motion is satisfactory for a ballasted cargo boat (Figure 1), a whale-catcher boat (Figure 2), and a Coast Guard cutter (Figure 3). Generally those ships with relatively large rudder sizes (usual merchant ships in ballasted condition are also included in this group) are appropriate for the present approach because of their small T values. Another result for a full-loaded tanker (Figure 4) is less favorable, although it may still be within a permissible tolerance. Disagreement of such an order is found commonly for a number of full-loaded tankers and sometimes for full-loaded cargo boats. It may be related to their large T values and relatively considerable speed reduction with steering. Adaptability of the present approach for most cargo boats half and full-loaded is between the one for ballasted ship and full-loaded tankers, as is shown in Figures 5 and 6. Another rather

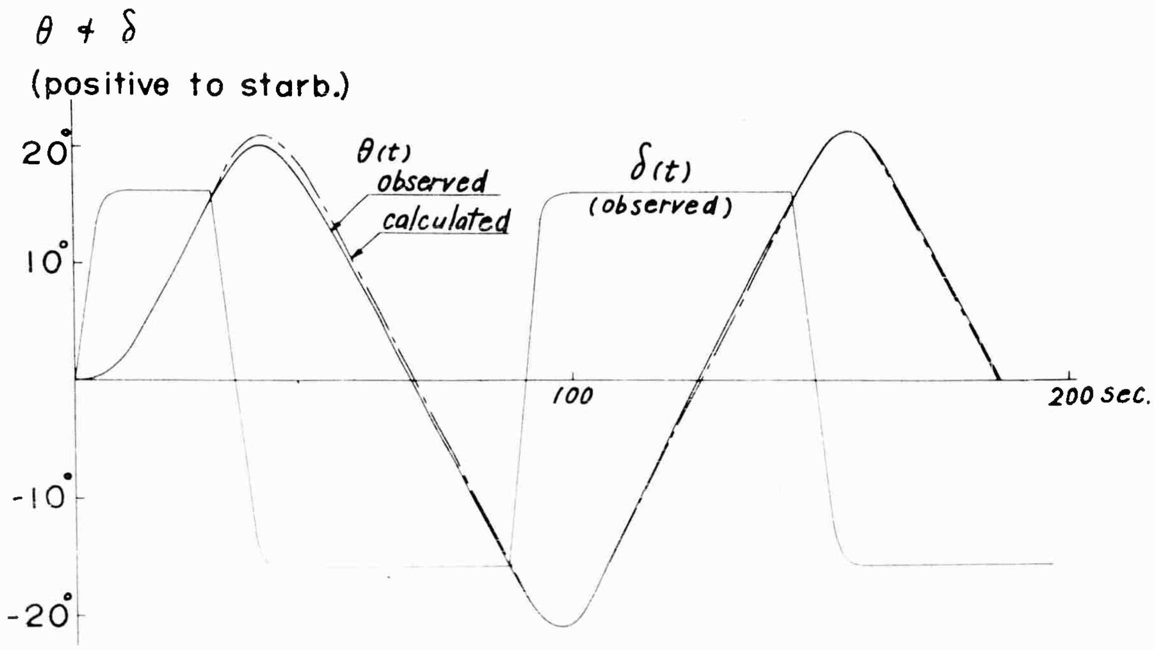

Figure 1 – Standard Maneuver Test for a Ballasted Cargo-Boat

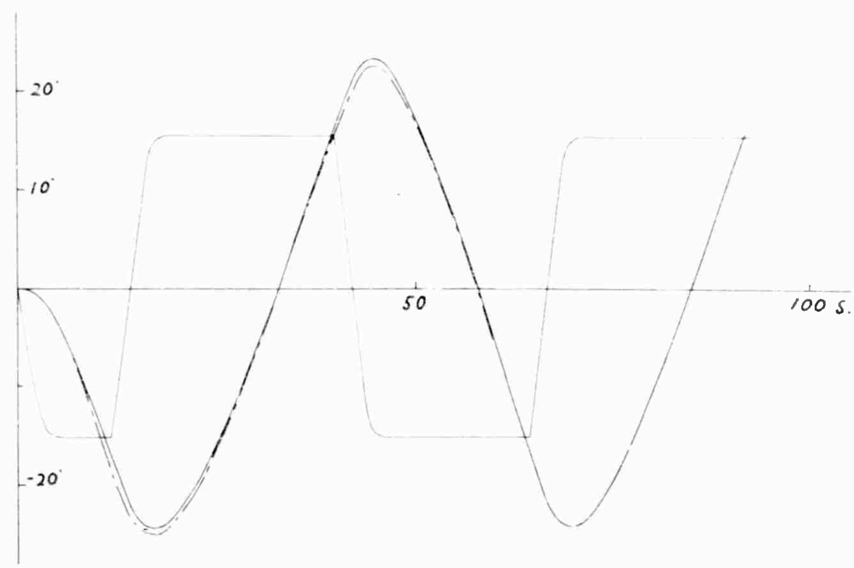

Figure 2 – Standard Maneuver Test for a Whale Catcher

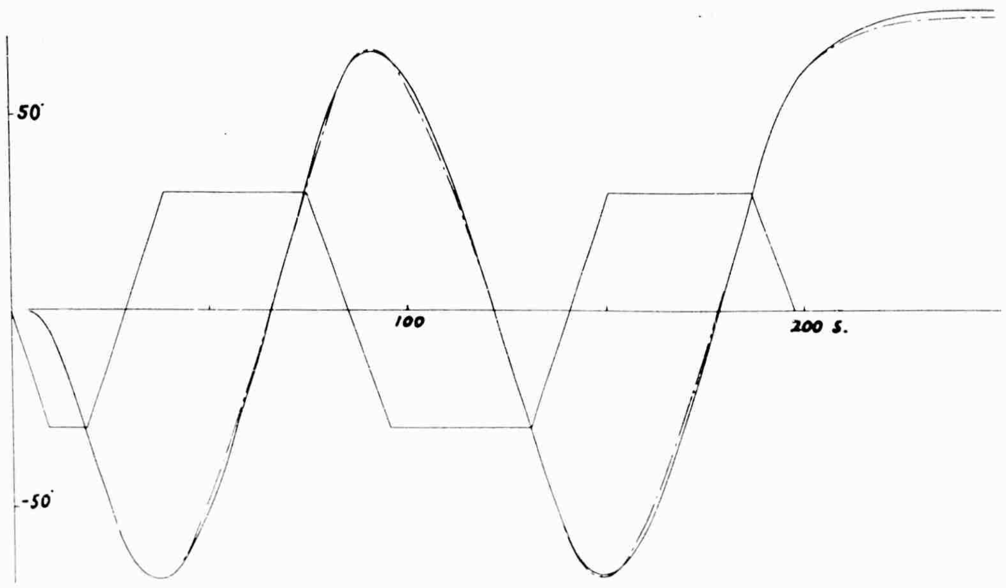

Figure 3 – Standard Maneuver Test for a Coast-Guard Cutter

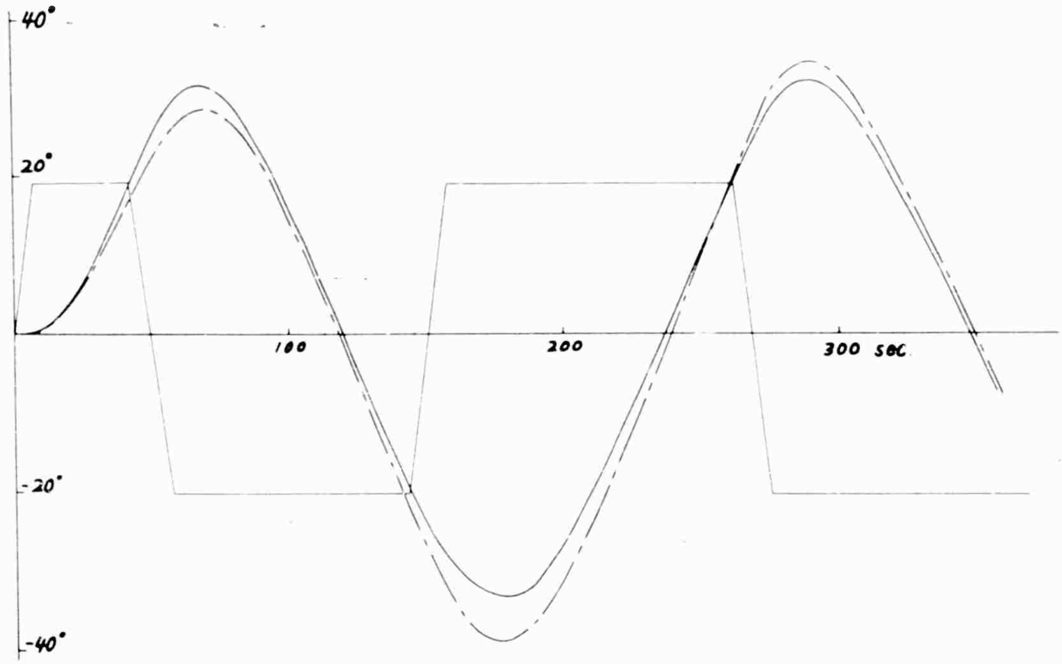

Figure 4 – Standard Maneuver Test for a Full-Loaded Tanker

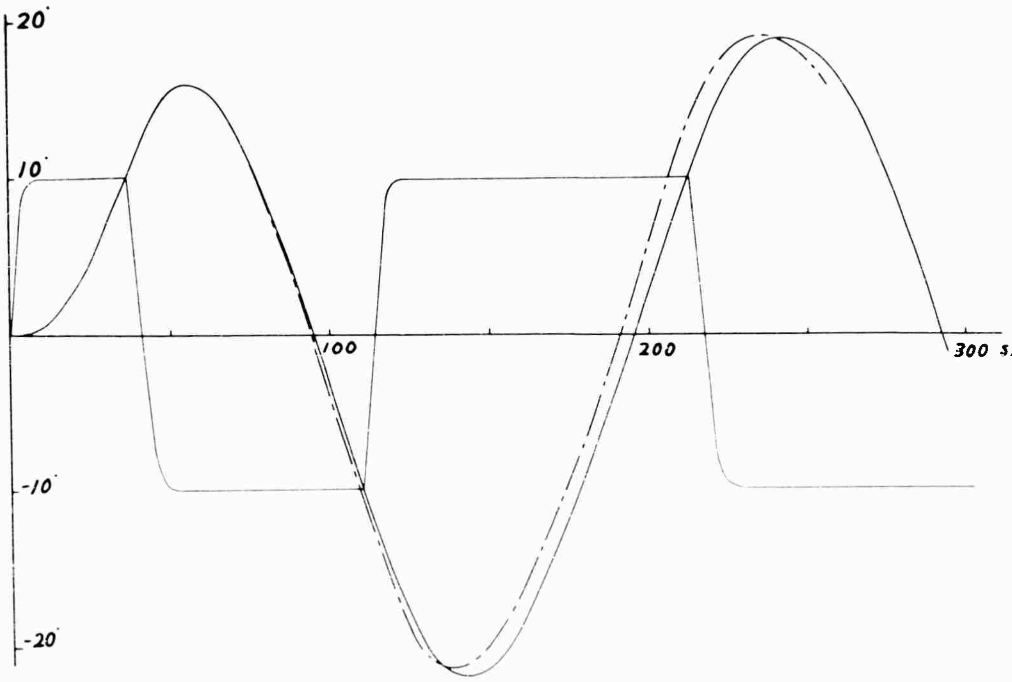

Figure 5 – Standard Maneuver Test for a Full-Loaded Cargo Boat

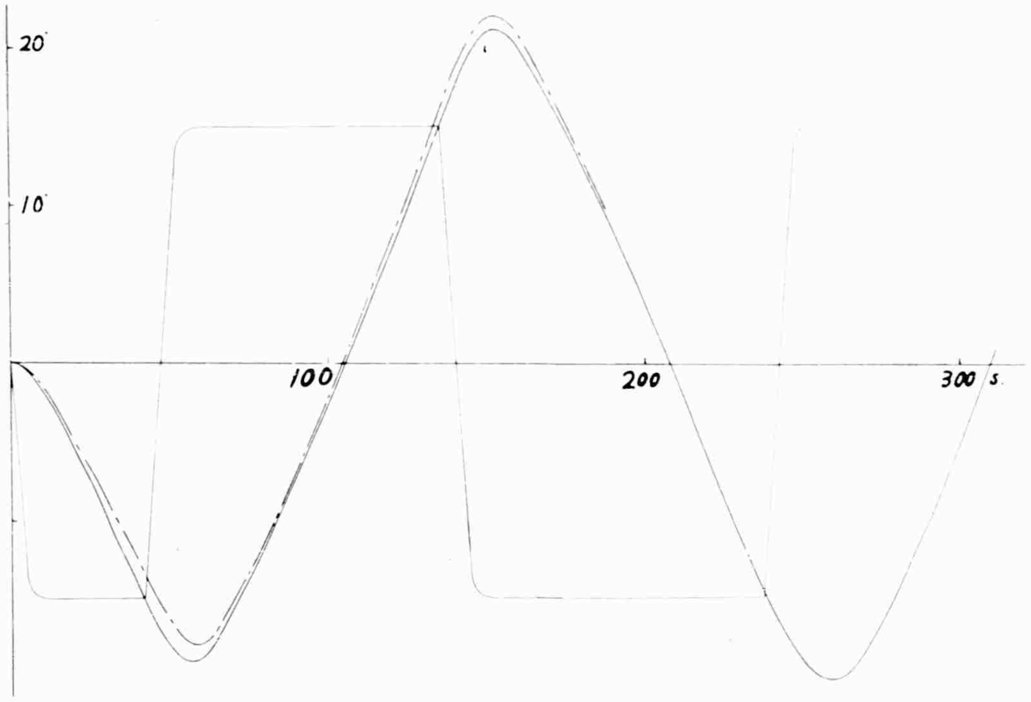

Figure 6 – Standard Maneuver Test for a Half-Loaded Cargo Boat

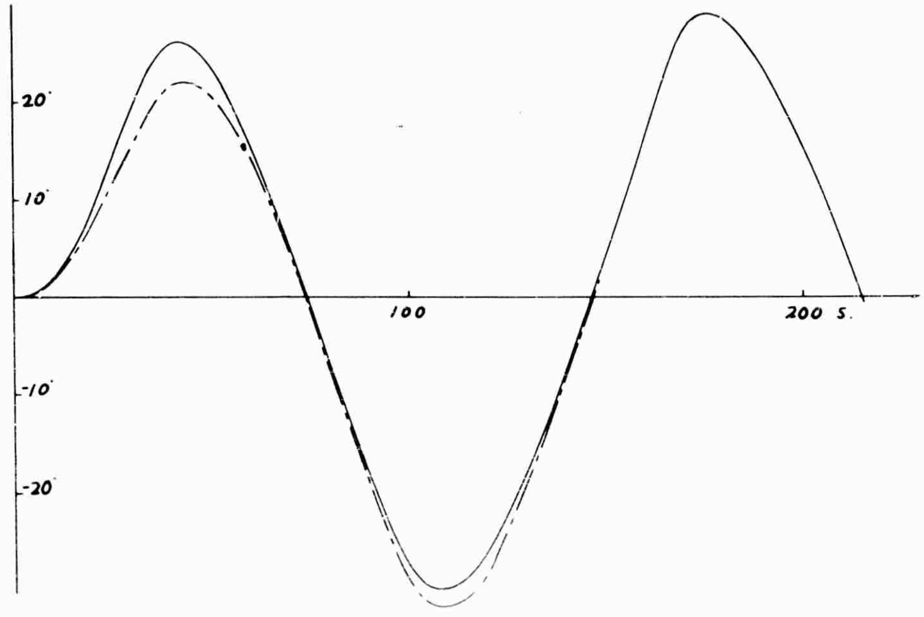

Figure 7 – Standard Maneuver Test for a Train Ferry

exceptional case is the result for a train-ferry boat with twin screws and twin rudders (Figure 7).

A survey of these results indicates that the present approach employing the first-order equation of motion may be satisfactory to describe steering motion of a ship as it has been predicted theoretically.[2]

1.3 REPRESENTATION OF MANEUVERABILITY IN TERMS OF PROPOSED STEERING QUALITY INDICES

The conclusion that motion of a ship in steering may be described essentially by the first-order Equation [2] leads us to the understanding that the dynamic character of a ship in steering (viz, so-called steering quality or maneuverability) is composed essentially of two elements: one is turning ability (represented by K) and the other is quick response in steering (represented by T). If a helm angle δ_0 is put on suddenly, the turning angular rate of a ship increases gradually and terminally approaches $K\delta_0$, as is obtained easily by solving Equation [2] as follows,

$$\dot{\theta}(t) = K\delta_0 \,(1 - e^{-t/T}) \qquad [3]$$

where $\delta = \delta_0$ for $t \geqq 0$, and

$\delta = 0$ for $t < 0$.

The index K indicates a ratio of a steady turning angular rate to a corresponding helm angle and may be called the index of turning ability; the larger the K value of a ship is, the greater

is the turning angular rate to which she approaches and then the smaller is her steady turning circle. On the other hand, the rapidity with which a ship approaches the terminal angular rate $K\delta_0$ is defined by the index T, as is obvious in the solution [3] of Equation [2]. Thus T may be called the index of quick response in steering; the smaller the T value of a ship, the quicker the decay of $e^{-t/T}$, and consequently also the quicker the buildup of her turning angular motion is.

Considering that actual maneuvering is a ceaseless succession of random steerings to starboard and to port, a quick response of a ship to steering is quite important for timely and swift maneuvering. Quick response and a small turning circle do not necessarily accompany each other but constitute two different abilities which are clearly distinct. Then it should be emphasized that maneuverability must be expressed not merely by the smallness of steady turning circle (greatness of K) but also by the quick response in steering (smallness of T) because maneuverability depends on a rapid display of an inherent turning rate as well as on the greatness of the inherent turning rate.

It may be an opportune approach here to visualize the present representation of maneuverability in terms of K and T, employing several typical combinations of these values. Let us take the combinations as follows:

	Ship "A"	Ship "B"	Ship "C"	Ship "D"	
K	0.050	0.050	0.065	0.065	1/sec
T	30	50	30	50	sec

Ship "A" corresponds to 20,000-DW Tanker, and other combinations are constructed by variation of K and T values. Patterns of buildup of turning angular rate and turning paths for these cases are illustrated in Figures 8 and 9. A combination of large K and small T means a quick buildup of a powerful turning, and this is just the feature of superior maneuverability, because a ship with such a character ("C") may outdo other ships in all phases of steering. Whereas a ship with both smaller K and T (Ship "A") is distinguished in the earlier phases by her quick buildup of turning rate, after awhile her rival with both larger K and T (Ship "D") overtakes and, displaying great turning ability, finally leaves Ship "A" behind. It should be noted that "A," a ship with both smaller K and T, outdoes her rival "D" so far as the earlier phase of motion is concerned, in spite of "A's" poor turning ability; that means a larger steady turning circle. This evidently depends upon the fact that Ship "A" can display her inherent turning ability more quickly although the inherent turning rate is not so large. Considering the transient figures of actual maneuvers, it is difficult to decide whether Ship "A" or Ship "D" has a more desirable maneuverability merely concerning their steady turning circle. In these circumstances, the present representation of maneuverability using the indices K and T may be reasonable.

The quick response in steering has been sometimes represented by "reach" or "turning lag," which are defined in Figure 10. These quantities may be written simply in terms of T as follows:

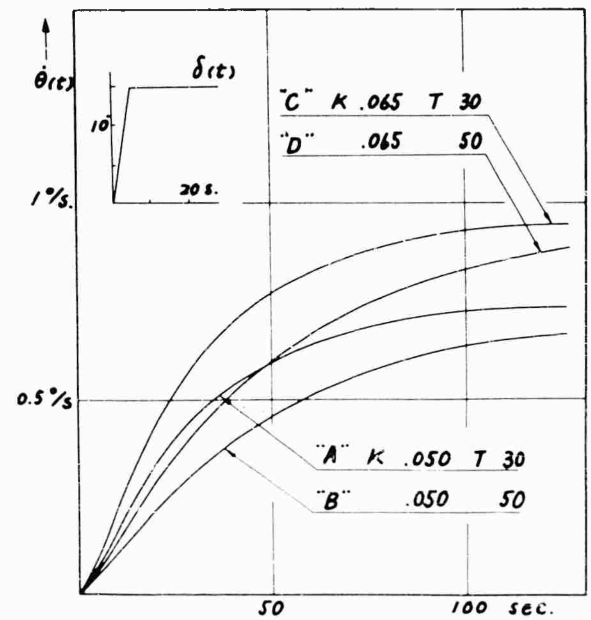

Figure 8 – Buildup of Turning Rate for Ships A, B, C, and D

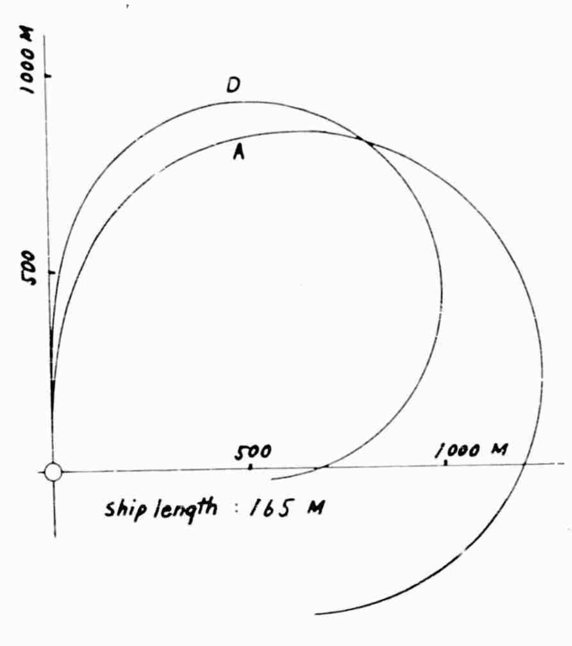

Figure 9 – Turning Paths of Ships A and D by a Helm Angle of 15 Degrees

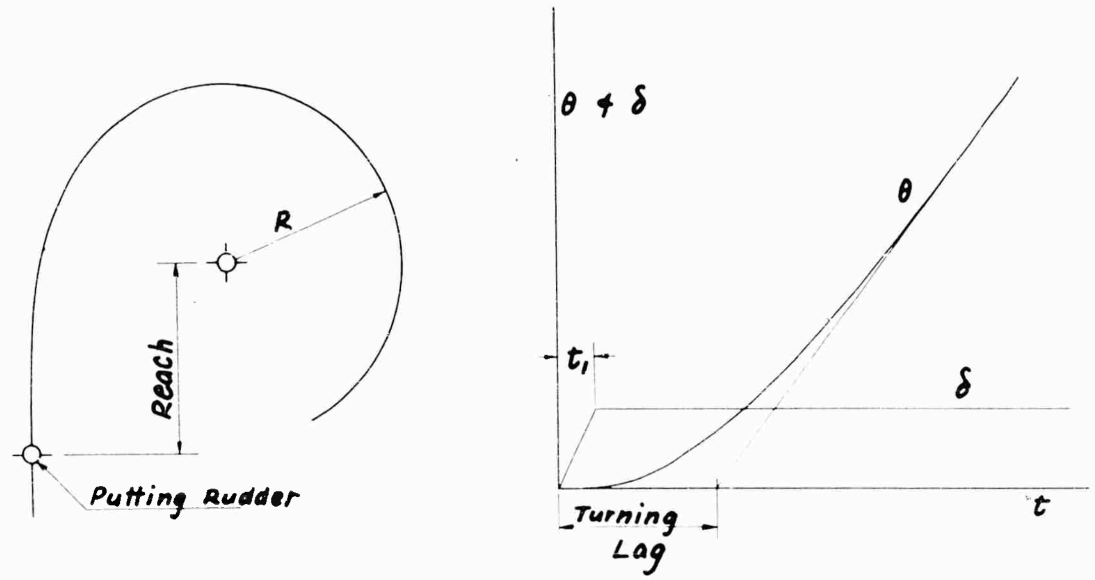

Figure 10 – Reach and Turning Lag

$$\text{turning lag} = T + \frac{t_1}{2} \approx T, \text{ and}$$

$$\text{reach} \qquad = V\left(T + \frac{t_1}{2}\right) \approx V \cdot T,$$

where t_1 is the time spent to set a helm angle (it is usually fairly small comparing T) and V is the ship speed.

On the other hand, steady turning radius R may be expressed in terms of K; that is,

$$R = V/K\delta_0$$

where δ_0 is the helm angle used, in radians. Then in addition,

$$\text{advance} \approx V\left(T + \frac{t_1}{2}\right) + \frac{V}{K\delta_0} \approx V \cdot T + \frac{V}{K\delta_0}$$

Thus turning radius and "reach" or "turning lag" represent, respectively, the turning ability and quick response, just as do the indices K and T. Then it is possible to represent maneuverability by steady turning radius R and "reach" or "turning lag" in place of K and T. To use these particulars may be sometimes convenient because of their extensive popularity. It should be added, however, that the present indices K and T have a unique utility in predicting its behavior of a ship in steering. If the values of the indices for a ship are given, her maneuvering motion for an arbitrary form of steering may be predicted, using the Equation of Motion [2]. The reliability of the procedure is already indicated by the analyses of the standard maneuver shown in Figures 1 to 7.

Finally, let us discuss so-called "stability on course" and a representation of it in terms of the present maneuvorability index. This quality of a ship relates to her behavior after she is disturbed by an external force and deviates from a straight-running course. According to experience, the ease of course-keeping depends largely upon this quality. A rate of heading deviation of most ships decays even with their rudders amidship after the external disturbance is removed, and finally they settle in a straight running but along a new direction somewhat different from the original course. A ship with such a character is called "stable on course." There is sometimes, however, an exceptional ship whose heading deviation rate does not decay after the removal of the external disturbance. Such a ship is said to be "unstable on course." Experiments show that instability and a very low degree of stability cause considerable difficulties in course-keeping.

The degree of stability on course may be expressed by the rapidity of the decay of heading deviation rate after the removal of a disturbance; the quicker the decay, the higher the stability. The Equation of Motion [2] indicates that the heading deviation rate of a ship with a rudder amidship after the removal of a disturbance is described as

$$\dot{\theta}\ (t) = \dot{\theta}_0\ e^{-t/T}$$

where $\dot{\theta}_0$ is the heading rate caused by the disturbance.
Then evidently the degree of stability on course depends on the rapidity of the decay of $e^{-t/T}$, which expresses just the degree of quick response in steering. Thus the stability on course agrees with the quick response in steering, and consequently the index of quick response T is also that of stability on course. The smaller the T value, the more stable is a ship on course.

A number of theoretical studies on steering of ships have dealt with stability on course, and most of them have given their stability indices. These indices and the present index T agree with each other in essential meaning, excluding some notational differences. For instance, Davidson's stability index p_1 (Reference 4) may be written in terms of the present paper as follows:

$$p_1 = -\left(\frac{L}{V}\right)\frac{1}{T_1} \approx -\left(\frac{L}{V}\right)\frac{1}{T} = -\frac{1}{T'}$$

where V is the ship speed;

L is the ship length;

T' is the index T in a nondimensional form; and where T_1 has been introduced in Section 1.1 briefly.

In summary, it may be concluded that maneuverability of a ship is reasonably represented by both the index of turning ability K and the index of quick response in steering and stability on course T, and that the behavior of a ship in steering may be predicted immediately, using these two indices.

1.4 PHYSICAL INTERPRETATION OF THE STEERING QUALITY INDICES

A survey of the form of the first-order Equation of Motion [2] leads us to understand the physical constitution of steering motion of a ship as follows:

a. A ship has inertia to resist a turning angular acceleration;

b. Hydrodynamic forces acting upon a hull and rudder induce a damping moment to resist a turning angular velocity proportional to the angular velocity;

c. When putting on a helm angle, a moment to produce a turning motion is induced in proportion to the helm angle.

Describing this constitution in the form of an equation of motion, we get

$$I_e \frac{d\dot{\theta}}{dt} + N\dot{\theta} = M\delta$$

Then the index T indicates a ratio of the inertia I_e to the damping moment coefficient N, and the index K indicates a ratio of the turning moment coefficient M to the damping moment coefficient N.

It should be noted, however, that a turning angular motion of a ship is necessarily coupled by a side-drifting motion. Consequently the inertia to resist turning acceleration is not merely a moment of inertia of a ship (including additional mass) but is composed also of her lateral mass. In the same manner, the damping moment acting upon a hull and rudder is composed of a hydrodynamic moment caused by a turning angular motion and another one caused by a drifting motion accompanied with the angular motion. Circumstances are similar also for the turning moment produced by steering. In these circumstances the foregoing inertia of a ship, the damping moment against angular motion, and the turning moment by a rudder are not the ones for a pure rotational motion but "equivalent" ones considering the coupling between an angular motion and a dritting motion. However the meaning of the term "equivalent" seems ambiguous in a strictly analytical sense, it is clear, however, that steering motion of a ship may be considered apparently as a pure rotational motion through using the equivalent inertia, damping moment, and turning moment, because the motion of a ship in steering can be described essentially by the first-order Equation of Motion [2]. This is the fundamental concept of the first-order simulation for steering motion of a ship, and its utility is to simplify the treatment so as to yield plain and essential descriptions of maneuverabilities of ships.

2. DETERMINATION OF THE STEERING QUALITY INDICES USING KEMPF'S STANDARD MANEUVER TEST

2.1 PROCEDURE OF ANALYZING THE STANDARD MANEUVER TEST TO OBTAIN THE STEERING QUALITY INDICES

The principle of the analysis is to find those values of the indices K and T with which

the Equation of Motion [2] may describe an observed ship motion in the test. The notation used here is shown in Figure 11.

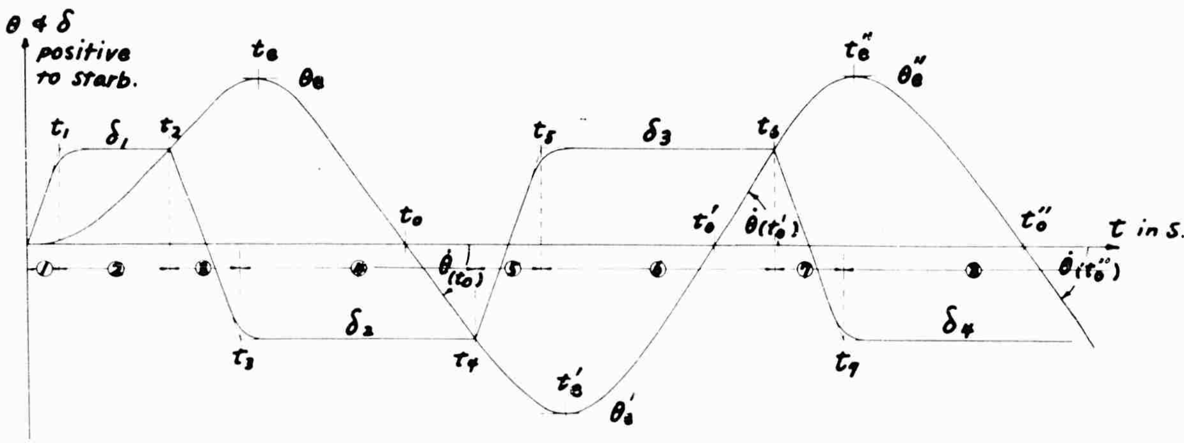

Figure 11 – Notations for Analysis of Kempf's Standard Maneuver

Numbers indicate successive periods during a test.

In taking up the analysis, it should be noted that a ship usually does not keep a straight course with a rudder apparently amidship (viz, shown by a helm indicator) but makes some slow turning by reason of an unsymmetric velocity field of single-screw race, small missetting of a rudder, and other miscellaneous factors. Since the equation of motion requires naturally a straight running for a rudder amidship, it is necessary to make some correction upon an apparent (observed) helm angle before putting it into Equation [2]. Then we put

$$\delta(t) = \delta m(t) + \delta r \qquad [4]$$

where $\delta m(t)$ is the observed helm angle, and

δr is the "residual helm," that is, the difference between $\delta(t)$ and $\delta m(t)$, and may be considered as an unknown constant at the beginning of the analysis.

Then putting Equation [4] into Equation [2], we obtain

$$T \frac{d\dot{\theta}}{dt} + \dot{\theta} = K\delta r + K\delta m(t)$$

Integrating both sides from $t = 0$ to $t = t$,

$$T \int_0^t \frac{d\dot{\theta}}{dt}\, dt + \int_0^t \dot{\theta}\, dt = K\delta r \int_0^t dt + K \int_0^t \delta m(t)\, dt$$

Setting the time origin at the start of a test, measuring $\theta(t)$ from the base course, and provided the ship is running straight before the test begins, the initial conditions are

$$\theta = 0 \text{ and } \dot{\theta} = 0 \text{ at } t = 0.$$

Then we get

$$T\dot{\theta} + \theta = K\delta r t + K\int_0^t \delta m(t)\, dt \tag{5}$$

This is the fundamental equation for the present procedure.

Applying Equation [5] at $t = t_e'$ and $t = t_e''$, we get

$$K\delta_r t_e' + K\int_0^{t_e'} \delta m(t)\, dt = \theta_e'$$

$$K\delta_r t_e'' + K\int_0^{t_e''} \delta m(t)\, dt = \theta_e''$$

since $\dot{\theta} = 0$ at these instants.

$\int_0^{t_e' \text{ and } t_e''} \delta m(t)\, dt$ may be obtained through a simple calculation indicated later. (Refer to Section 2.2.) Then unknown quantities in these simultaneous equations are K and δ_r. They may be determined by resolving these equations simultaneously. We call that K, *K in after-half period in a test*, writing simply $K_{⑥ ⑧}$.

Now applying Equation [5] at $t = t_e$, we get

$$K = \frac{\theta_e}{\delta_r t_e + \int_0^{t_e} \delta m(t)\, dt}$$

δ_r has been determined through the above procedure, and $\int_0^{t_e} \delta m(t)\, dt$ may be calculated. (Refer to Section 2.2.) We call this K, *K in earlier-half period in a test*, writing simply $K_{④}$.

$K_{④}$ and $K_{⑥ ⑧}$ are more or less different from each other by reason of speed reduction in a test and nonlinear effect for steering quality indices. Usually it may be considered that the mean value of $K_{④}$ and $K_{⑥ ⑧}$ is a representative K of a ship that is tested.

Next, applying Equation [5] at $t = t_0$, $t = t_0'$ and $t = t_0''$, we get

$$T = \frac{K}{\dot{\theta}(t_0)}\left(\int_0^{t_0} \delta m(t)\, dt + \delta_r t_0\right)$$

$$T = \frac{K}{\dot{\theta}(t_0')} \left(\int_0^{t_0'} \delta m(t)\, dt + \delta_r t_0' \right)$$

$$T = \frac{K}{\dot{\theta}(t_0'')} \left(\int_0^{t_0''} \delta m(t)\, dt + \delta_r t_0'' \right)$$

since $\theta = 0$ at these instants. When carrying out the calculations, it may be reasonable to use $K_{④}$ for the first formula and to use $K_{⑥\,⑧}$ for the remaining two. The values for $\dot{\theta}(t_0)$, $\dot{\theta}(t_0')$, and $\dot{\theta}(t_0'')$ may be determined from the plot of $\theta(t)$.

The integral $\int_0^{t_0,\ t_0' \text{ and } t_0''} \delta m(t)\, dt$ may be calculated. (Refer to Section 2.2.) We call the T obtained from the first formula T *in earlier-half period*, writing simply $T_{④}$, and call the average of T's from the remaining two formulas T *in after-half period* simply $T_{⑥\,⑧}$. The mean value of $T_{④}$ and $T_{⑥\,⑧}$ may be considered as a representative T of a ship that is tested.

2.2 CALCULATING TABLE FOR PRACTICING THE ANALYSIS

A trapezoidal approximation is quite adequate for $\delta(t)$ produced by usual steering gears; the error of it does not exceed other unavoidable errors. Then we get formulas for calculating $\int_0^t \delta m(t)\, dt$ as follows:

No. of period

① $\displaystyle \int_0^t \delta m(t) = \delta_1 \frac{t^2}{2t_1}$

② $\displaystyle = \delta_1 \left(t - \frac{t_1}{2} \right)$

③ $\displaystyle = \delta_1 \left(t - \frac{t_1}{2} \right) + \frac{\delta_2 - \delta_1}{2(t_3 - t_2)} (t - t_2)^2$

④ $\displaystyle = \delta_1 \left\{ -\frac{t_1}{2} + \frac{1}{2}(t_2 + t_3) \right\} + \delta_2 \left\{ -\frac{1}{2}(t_2 + t_3) + t \right\}$

⑤ $\displaystyle = \delta_1 \left\{ -\frac{t_1}{2} + \frac{1}{2}(t_2 + t_3) \right\} + \delta_2 \left\{ -\frac{1}{2}(t_2 + t_3) + t \right\} + \frac{\delta_3 - \delta_2}{2(t_5 - t_4)} (t - t_4)^2$

No. of period

(6)
$$\int_0^t \delta m(t) = \delta_1 \left\{ -\frac{t_1}{2} + \frac{1}{2}(t_2+t_3) \right\} + \delta_2 \left\{ -\frac{1}{2}(t_2+t_3) + \frac{1}{2}(t_4+t_5) \right\} + \delta_3 \left\{ -\frac{1}{2}(t_4+t_5) + t \right\}$$

(7)
$$= \delta_1 \left\{ -\frac{t_1}{2} + \frac{1}{2}(t_2+t_3) \right\} + \delta_2 \left\{ -\frac{1}{2}(t_2+t_3) + \frac{1}{2}(t_4+t_5) \right\} + \delta_3 \left\{ -\frac{1}{2}(t_4+t_5) + t \right\} + \frac{\delta_4-\delta_3}{2(t_7-t_6)}(t-t_6)^2$$

(8)
$$= \delta_1 \left\{ -\frac{t_1}{2} + \frac{1}{2}(t_2+t_3) \right\} + \delta_2 \left\{ -\frac{1}{2}(t_2+t_3) + \frac{1}{2}(t_4+t_5) \right\} + \delta_3 \left\{ -\frac{1}{2}(t_4+t_5) + \frac{1}{2}(t_6+t_7) \right\} + \delta_4 \left\{ -\frac{1}{2}(t_6+t_7) + t \right\}$$

When carrying out the analysis following the above procedure, it is convenient to utilize a calculating table. Table 1 shows such a table as to suit those ships which have t_e in the period (4), t_e' in (6), and t_e'' in (8), as this is the case for most ships. If a ship that is tested does not satisfy this condition, the table must be modified partially, according to the above formulae, obtaining $\int_0^t \delta m(t)\, dt$.

2.3 RESULTS OF THE ANALYSES ON THE STANDARD MANEUVER TESTS FOR APPROXIMATELY SEVENTY ACTUAL SHIPS

The Osaka University Tank has been collecting the standard maneuver results for actual ships at sea and analyzing them following the present procedure. Table 2 illustrates all reliable data obtained up to the present. The notations used in the table are shown below To construct a systematic knowledge on the steering quality indices K and T, many more similar data are desirable as well as systematic research employing self-propelled, free-running models.

TABLE 1

Calculating Table for Analysis of Standard Maneuver Test

Ship Antoinette

④	t_e	72.9	θ_e	33.7	t_0	131.9	$\dot{\theta}_{t_0}$	−0.917	t_1	6.0	t_5	166.2	δ_1	21.0
⑥	t_e'	187.5	θ_e'	−36.0	t_0'	261.0	$\dot{\theta}_{t_0}'$	0.833	t_2	46.9	t_6	285.9	δ_2	−20.6
⑧	t_e''	317.9	θ_e''	34.5	t_0''	351.3	$\dot{\theta}_{t_0}''$	0.820	t_3	58.9	t_7	297.9	δ_3	20.8
									t_4	154.2			δ_4	−20.8

t_1	6.0	t_2	46.9	t_4	154.2	t_6	285.9
$t_1/2$	3.0	t_3	58.9	t_5	166.2	t_7	297.9
		(t_2+t_3)	105.8	(t_4+t_5)	320.4	(t_6+t_7)	563.8
		$(\ )/2$	52.9	$(\ )/2$	160.2	$(\ )/2$	291.9

④		⑥		⑧	
				$\delta_1(\ )$	1048
		$\delta_1(\ )$	1048	$\delta_2(\ )$	−2210
$-\frac{t_1}{2}$	−3.0	$-(t_2+t_3)/2$	−52.9	$-(t_4+t_5)/2$	−160.2
$(t_2+t_3)/2$	52.9	$(t_4+t_5)/2$	160.2	$(t_6+t_7)/2$	291.9
$(+)$	49.9	$(+)$	107.3	$(+)$	131.7
$\delta_1(+)$	1048	$\delta_2(+)$	2210	$\delta_3(+)$	2746
$-(t_2+t_3)/2$	−52.9	$-(t_4+t_5)/2$	−160.2	$-(t_6+t_7)/2$	−291.9
t_e	72.9	t_e'	187.5	t_e''	317.9
$(+)$	20.0	$(+)$	27.3	$(+)$	26.0
$\delta_2(+)$	−412	$\delta_3(+)$	568	$\delta_4(+)$	−541
$\int_0^{t_e} \delta\, dt$	636	$\int_0^{t_e'} \delta\, dt$	−594	$\int_0^{t_e''} \delta\, dt$	1037
(t_0-t_e)	59.0	$(t_0'-t_e')$	73.5	$(t_0''-t_e'')$	63.4
$\delta_2(\ )$	−1215	$\delta_3(\ )$	1528	$\delta_4(\ )$	−1319
$\int_0^{t_0} \delta m\, dt$	−579	$\int_0^{t_0'} \delta\, dt$	934 4	$\int_0^{t_0''} \delta\, dt$	−282
$\delta_r t_0$	−124	$\delta_r t_0'$	−245 5	$\delta_r t_0''$	−358
$(+)$	−703	$(+)$	689 9	$(+)$	−640
$\dot{\theta}_{t_0}$	−0.917	$\dot{\theta}_{t_0}'$	0.833	$\dot{\theta}_{t_0}''$	−0.820
T/K	767		827		780
K	0.0593		804		827
T ④	45.5		0.0467		1607
			37.6		804
		T ⑥⑧	45.5		
K_{mean}	0.0530		83.1		
T_{mean}	41.6		41.6		

	t_e'		$\int^{t_e'} \delta\, dt$		θ_e'
⑥	$187.5 \times K\delta_r$	−	594	$\times K$	$= -36.0$
⑧	317.9 ··	+	1,037	··	$=$ 34.5
⑥	× 317.9	−	188,833		−11,444
⑧	× 187.5		194,438		6,469
			383,271	K	= 17,913
				K ⑥⑧	0.0467
⑥	$187.5 \times K\delta_r$	−	594	× 0.0467	$= -36.0$
				−27.75	27.75
					−8.25
					−0.0440
⑧	$317.9 \times K\delta_r$	+	1,037	× 0.0467	= 34.5
					−48.43
					−13.93
					−0.0438
	④			$K\delta_r$	= −0.0439
δ_r			−0.939	δ_r	= −0.939
t_e			72.9		
$\delta_r t_e$			−68.5		
$\int_0^{t_e} \delta m\, dt$			636		
$(+)$			568		
θ_e			33.7		
K ④ $= \theta_e/(+)$			0.0593		
K ⑥⑧			0.0467		
			0.1060		
K_{mean}			0.0530		

TABLE 2

Standard Maneuver Test Results for Approximately Seventy Actual Ships

Kind of Ship	Condition	$L \times B \times D$	d_a	d_f	d	Δ	V_{kt}	∇/L^2d	A_R/Ld	δ_0	K	T	K'	T'
C	B	152 × 20.6 × 12.7	5.61	2.44	4.02	8,828	17.2	0.093	1/ 36.3	+ 15 − 20	0.043 0.041	11 10	0.74 0.70	0.64 0.58
C	B	114 × 16.4 × 9.3	4.58	1.96	3.27	4,180	15.7	0.096	1/ 30.2	+ 15	0.054	6.9	0.76	0.49
C	B F	145 × 19.5 × 12.2	8.78	7.26	8.02	15,780	18 14.8	0.091	1/ 64.6	20 20	0.043 0.052	7.0 25	0.67 0.98	0.45 1.3
C	B	132 × 18.2 × 11.7	5.10	2.01	3.55	5,966	17.1	0.096	1/ 32.8	20	0.057	8.2	0.86	0.55
C	B	115 × 16.3 × 9.25	4.07	1.98	3.02	4,218	16.1	0.105	1/ 24.8	15 − 15	0.053 0.061	5.4 6.3	0.74 0.85	0.39 0.46
C	B	137 × 18.5 × 8.95	5.03	2.18	3.61	6,430	16.8	0.093	1/ 35.3	10 15	0.067 0.070	10.0 11.0	1.07 1.11	0.63 0.69
C	F	144 × 19.3 × 9.50	8.54	7.98	8.26	15,573	17.1	0.089	1/ 65.8	+ 10 − 10	0.088 0.084	54.8 47.6	1.45 1.37	3.35 2.91
C	F	157 × 19.6 × 12.5	8.60	7.90	8.25	16,000	17.1	0.077	1/ 69.6	+ 10 − 10	0.071 0.073	26.2 26.7	1.27 1.30	1.47 1.50
C	H	121 × 16.2	6.36	3.94	5.15	6,750	10	0.087		+ 15 − 15	0.040 0.035	18.0 17.0	0.94 0.82	0.76 0.72
C	F	133 × 18.6 × 10.4	8.62	7.58	8.10	15,160	14.3	0.103	1/ 65.2	+ 10 − 10	0.102 0.085	53.2 46.9	1.84 1.54	2.94 2.59
C	F	148 × 19.4 × 12.30	8.30 9.04	7.58 7.77	7.94 8.40	15,305 16,350	17.5 17.5	0.046 0.050	1/ 59.6 1/ 63.1	− 10 + 15 − 15	0.070 0.063 0.060	21.1 22.5 22.6	1.15 1.03 0.98	1.29 1.37 1.38
C	B	138 × 18.8 × 11.85	5.86	2.06	3.96	6,972	17.0	0.090	1/ 28.3	10	0.061	11.3	0.97	0.72
C	F	123 × 16.7 × 9.45	7.00	6.30	6.65	10,623	9.4	0.104	1/ 63.2	10	0.080	85.2	2.02	3.36
O.C.	B	136 × 20.4	6.42	4.56	5.49	10,899	12	0.105		15 − 15	0.048 0.046	22.1 25.0	1.06 1.02	1.00 1.14
C	F	129 × 18.2 × 11.1	8.73	8.13	8.43	15,030	8.8	0.094	1/ 64.9	10 − 10	0.042 0.040	50 42	1.20 1.14	1.75 1.47
C	H	115 × 16.3 × 9.0	5.46	4.44	4.95	6,470	12.0	0.096	1/ 40.4	− 10	0.084	30.4	1.56	1.63
C	F	123 × 16.5 × 9.0	7.52	7.04	7.28	10,391	11.3	0.092	1/ 63.7	− 10	0.059	30.4	1.24	1.44
C	1/3 L	123 × 16.3	4.20	2.64	3.42		11.0			15 − 15	0.050 0.045	30 30.4	1.09 0.97	1.38 1.39
C	F	137 × 18.5	8.30	7.84	8.07	14,407	16.0	0.094	1/ 66.2	− 15	0.073	30.9	1.20	1.86
C	L	114 × 16.2					11.5			15	0.047	10	0.91	0.52
C	F	140 × 19 × 10.5	8.70	8.00	8.35	16,050	15.0	0.096	1/ 59.9	10	0.094	35	1.70	1.93
C	B	106 × 15.6 × 8.1	4.30	1.77	3.04	3,275	13.0	0.094	1/ 28.4	10	0.067	8.3	1.06	0.53
C	F	86 × 12.5 × 5.5	5.80	5.53	5.66	4,493	8.5	0.106	1/ 64.2	10 − 10	0.102 0.109	36.6 44.2	2.00 2.14	1.87 2.26
C	B	128 × 17.8 × 10.4	5.39	2.48	3.94	6,188	13.0	0.127	1/ 31.9	10 − 10	0.058 0.059	11.6 16.4	1.11 1.12	0.61 0.86
Training Barque	H	85 × 13.4	5.37	4.19	4.78	3,080	9.0	0.086	1/ 40.4	15 − 15	0.039 0.041	18.6 19.6	0.73 0.76	1.01 1.06
C	F	156 × 19.5	7.50	6.85	7.18	13,770	16.5	0.077	1/ 62.2	15 − 15	0.062 0.070	25 27	1.14 1.28	1.36 1.47
C	H	94 × 13.7 × 7.6	5.26	3.04	4.15	3,800	10.3	0.101	1/ 34.2	10 − 10	0.059 0.059	20 19	1.04 1.04	1.18 1.08

TABLE 2 (Continued)

Kind of Ship	Condition	$L \times B \times D$	d_a	d_f	d	Δ	V_{kt}	∇/L^2d	A_R/Ld	δ_0	K	T	K'	T'
C	H	122 × 17.4	6.26	3.30	4.78	7,160	14	0.098		15 −15	0.050 0.055	11.6 11.4	0.85 0.93	0.68 0.67
C	F	112 × 15.8	7.63	7.00	7.32	9,670	11.5	0.103	1/ 71.8	−15	0.071	50.8	1.34	2.68
C	F	161 × 20.4 × 12.0	9.02	9.02	9.02	20,987	12.8	0.088	1/ 67.1	−10	0.053	47.1	1.30	1.92
C	F	145 × 18.3	8.00	7.30	7.65	14,000	14.0	0.085		15 15	0.090 0.067	69.2 43.4	1.82 1.36	3.43 2.20
C	F	145 × 19.0 × 12.2	7.95	7.12	7.54	14,400	15	0.089	1/ 60.8	10	0.073	29.7	1.36	1.58
C	F	141 × 18.0	7.10	6.63	6.86	11,800	13	0.084		15 −15	0.061 0.059	36.2 31.6	1.28 1.72	1.25 1.50
C	B	134 × 18.4	5.76	2.17	3.96	6,644	15.3	0.091	1/ 29.1	15 −15	0.051 0.049	8.8 8.1	0.87 0.84	0.52 0.48
C	F	122 × 15.9 × 9.45	7.94	7.30	7.62	11,580	9.5	0.100	1/ 57.4	10 −10	0.081 0.077	71.0 73.6	2.01 1.90	3.77 2.86
C	F	122 × 15.9 × 11.46	8.52	7.87	8.20	12,353	9.8	0.099	1/108.7	10 −10	0.056 0.061	73.6 81.2	1.36 1.49	3.03 3.34
C	F	112 × 16.2 × 9.0	7.84	7.00	7.42	9,980	11.5	0.105	1/ 64.2	10	0.119	47.9	2.25	2.53
C	F	140 × 18.2	8.06	7.59	7.82	13,750	12.0	0.088	1/ 75.2	15	0.085	76	1.12	2.08
T	F	201 × 28.2 × 14.6	10.85	10.81	10.83	50,700	17.6	0.113	1/75.8	20	0.058	39.5	1.28	1.78
T	F	193 × 26.5 × 13.87	10.27	10.35	10.31	42,910	17.0	0.110	1/72.4	20	0.067	56.2	1.48	2.55
T	F	192 × 26.5 × 13.87	10.28	10.35	10.32	43,080	16.5	0.110	1/72.3	20	0.052	54.4	1.18	2.40
T	F	193 × 26.5 × 13.87	10.32	10.31	10.31	43,182	17.2	0.110	1/73.2	20	0.050	48.3	1.08	2.22
T	F	193 × 26.5 × 13.87	10.31	10.33	10.32	42,920	16.5	0.109	1/72.5	20	0.049	48.5	1.11	2.13
T	F	185 × 25.2 × 13.4	10.50	10.10	10.30	37,695	15.5	0.104	1/75.4	10 10	0.086 0.063	95.0 42.4	2.00 1.46	4.09 1.83
T	B	167 × 21.5	5.95	3.65	4.80	12,800	13.5	0.093		15 −15	0.045 0.055	21.6 41.7	1.09 1.33	0.90 1.73
T	F	181 × 25.4 × 13.5	10.15	10.14	10.14	37,520	17.0	0.110	1/72.2	20	0.053	41.9	1.09	2.03
T	F	167 × 22.0 × 12.2	9.31	9.29	9.30	27,137	15.9	0.102	1/65.3	20	0.052	33.6	1.07	1.64
T	F	201 × 28.2 × 14.6	10.84	10.82	10.83	50,540	17.8	0.115	1/73.2	20	0.055	55.6	1.21	2.53
O.T.	B F	216 × 30.6 × 15.4	7.47 10.29 10.31	3.07 10.21 10.32	5.27 10.25 10.32	26,900 56,250 56,700	18.0 17.6 17.6	0.107 0.115 0.115	1/37.5 1/71.1 1/71.1	15 20 10	0.027 0.056 0.171	11.4 61.3 215	0.63 1.33 4.08	0.49 2.57 9.01
O.T.	F	216 × 30.6 × 15.4	10.44	10.43	10.44	57,100	17.2	0.115	1/69.4	20 35	0.050 0.040	68.7 58.1	1.22 0.97	2.82 2.38
T	B F	201 × 28.2 × 14.6	7.12 10.81	2.67 10.82	4.89 10.84	21,114 50,618	18.9 17.4	0.104 0.113	1/36.0 1/77.4	15 15 20	0.038 0.083 0.056	10.6 83.0 59.8	0.79 1.86 1.25	0.51 3.70 2.67
T	F	160 × 20	9.38	9.38	9.38	21,000	12.3	0.085	1/83.6	15 −15	0.050 0.043	43.9 32.3	1.27 1.08	1.73 1.27
T	F	192 × 26.5 × 13.87	10.43 10.61	10.43 10.60	10.43 10.60	43,100 44,090	16.0 16.1	0.109 0.110	1/72.1	10 15 −15	0.073 0.060 0.051	80.5 56.5 47.1	1.70 1.39 1.17	3.44 2.44 2.03
T	2/3 F	106 × 16.2 × 8.0	5.45	5.10	5.27	6,928	14	0.114	1/51.5	10 −10	0.110 0.103	41.6 34.6	1.62 1.52	2.83 2.35
T	F	154 × 20 × 11.5	8.85	9.18	9.02	20,583	12.3	0.094	1/71.8	10 −10	0.094 0.091	72.5 72.4	2.31 2.22	2.96 2.96

TABLE 2 (Continued)

Kind of Ship	Condition	$L \times B \times D$	d_a	d_f	d	Λ	V_{kt}	∇/L^2d	A_R/Ld	δ_0	K	T	K'	T'
T	B	167 × 22.0 × 12.2	7.05	3.80	5.43	15,025	16.0	0.097	1/38.6	15 −15	0.040 0.039	10 10	0.80 0.78	0.49 0.49
T	F B	192 × 26.8 × 13.7	10.60 6.44	10.20 1.85	10.40 4.14	43,000 13,000	15.0 16.0	0.109 0.083	1/74.2 1/29.6	10 10 −10	0.063 0.037 0.044	83.8 9.85 10.6	1.58 0.87 1.02	3.36 0.42 0.45
T	B	121 × 16.3	5.45	2.95	4.20	5,470	13.6	0.086		15 −15	0.053 0.053	10.2 12.9	0.92 0.93	0.59 0.74
T	B F	162 × 21.4	7.19 9.98	4.70 9.57	5.94 9.78	14,730 26,810	15.3 14.0	0.092 0.102	1/45.5 1/74.9	15 −15	0.057 0.086	12.6 75.3	1.21 1.94	1.17 3.35
W	Arrival	42 × 7.3 × 4.45	4.70	4.00	4.35		14.5		1/36.5	20 −20	0.193 0.173	5.85 5.05	1.09 0.97	1.04 0.90
W	Arrival	45 × 8.2 × 4.4	5.00	3.45	4.23	875	15.5	0.100	1/30.0	−15	0.185	7.6	1.04	1.35
W	Arrival	57 × 9.7 × 5.1	5.12	3.62	4.37	1,304		0.090	1/27.7	−15	0.199	5.2	1.29	0.80
W	Trial	57 × 9.7 × 5.1	4.56	2.34	3.45	935	17.4	0.081	1/21.8	−15	0.155	4.1	0.99	0.64
W	Arrival	57 × 9.5 × 5.1	4.65	2.27	3.52	889	17.7	0.076	1/21.2	−15	0.269	8.6	1.69	1.37
Emigrant Ship	1/2 F	145 × 20.4 × 11.9	7.20	5.50	6.35	12,070	19.0	0.018	1/49.7	15	0.053	11.8	0.78	0.80
Emigrant Ship	B	145 × 19.6	6.71	4.50	5.60	10,100	16.5	0.018	1/43.7	15 −15	0.060 0.066	19.2 18.4	1.03 1.12	1.12 1.08
Refrigerated Carrier	Trial	67 × 10.8 × 5.7	3.28	0.99	2.14	943	13.0	0.096	1/23.3	−15	0.070	5.1	0.70	0.51
C	Trial	41 × 8.2 × 3.75	[illegible]	0.59	1.42	270	11.1	0.111	1/19.5	15 −15	0.096 0.091	3.5 3.8	0.69 0.66	0.49 0.53
C	F	75 × 11.9 × 55	4.80	4.78	4.79	3,222	10.0	0.118	1/55.2	10 −10	0.077 0.093	25.4 22.9	1.1 1.4	1.75 1.58
		88 × 8.7 × 5.45	3.42	3.10	3.26	1,308	23.1 26.3	0.051	1/52.5	15 15 20 20	0.152 0.125 0.165 0.130	8.8 4.4 11.1 5.3	1.12 0.81 1.22 0.84	1.20 0.67 1.50 0.81
Train Ferry	H	111 × 17.4 × 6.80	4.92	4.64	4.78	5,370	14.4	0.089	1/30.1	15	0.096	22.6	1.44	1.51
Train Ferry	H	113 × 15.9 × 6.80	4.70	4.10	4.40	4,585	15.0	0.079	1/42.5	15	0.305	107	4.47	7.30
Coast-Guard Cutter	H	51.5 × 7.7 × 4.5	2.89	2.58	2.73	534	13.0	0.072	1/40	10 20 30	0.206 0.162 0.155	13.1 12.2 13.8	1.66 1.42 1.55	1.62 1.39 1.38

Kinds of Ships:
C is cargo boat. T is tanker. W is whale-catcher boat.

Condition at Test:
B is ballasted. F is full-loaded. H is half-loaded.

Particulars of Ships:
L is length between P.P. B is moulded breadth. D is depth. d_f, d_a, and d are draft at FP, AP and mean draft, respectively. (These dimensions are all in meters.)

Λ is displacement weight in metric tons. ∇ is displacement volume in cubic meters. A_R is rudder area in square meters.

V represents ship speed in meters per second. V_{kt} represents ship speed in knots.

Steering Quality Indices and Helm Angle Used:

$$K' = \left(\frac{L}{V}\right)K \qquad T' = \left(\frac{V}{L}\right)T$$

(This nondimensionalization is discussed in Section 3.1.)

δ_0 is used helm angle, positive to starboard, in degrees.

3. SYSTEMATIC FORMULATION OF THE STEERING-QUALITY INDICES AS FUNCTIONS OF HULL AND RUDDER PARTICULARS

It may be of both practical and theoretical interest to formulate the steering-quality indices as functions of hull and rudder particulars (viz, relative rudder sizes, slenderness factors of hulls, and so on) and then to provide an approach of predicting maneuverability of a ship when these particulars are given. While this attempt seems to be rather too ambitious to obtain complete success so easily, the use of the present indices K and T may be one of the most promising approaches. This chapter relates to such a formulation of the steering quality indices based on the analyses of Kempf's standard maneuver for actual ships illustrated in Section 2, sometimes referring to results of several self-propelled, free-running model tests. The formulation yields a rough estimation of the indices K and T and then a brief prediction of turning ability, quick response, and stability on course for usual merchant ships in full and half-loaded conditions. Similar formulation for other groups of ship types may also be provided in the same manner if adequate data for those ships are given.

3.1 NONDIMENSIONAL EXPRESSION OF THE INDICES K AND T

The physical interpretation of the indices, discussed in Section 1.4, indicates that:

$$K = \frac{M}{N} \quad \text{and} \quad T = \frac{I_e}{N},$$

where M is a coefficient of turning moment caused by steering and N is a coefficient of damping against turning motion, and where I_e represents the inertia of a ship, as has been discussed in detail.

A nondimensional representation of the damping moment $N\dot{\theta}$ may be written as

$$\frac{N\dot{\theta}}{\frac{\rho}{2} L^2 d V^2} = C_N \cdot \left(\frac{L}{R}\right)$$

where L is the ship length,
d is the mean draft,
V is the ship speed,
C_N is the nondimensional damping moment coefficient,
R is the instantaneous turning radius,

and then $\left(\frac{L}{R}\right)$ is the nondimensional turning angular velocity.

Considering that $R\dot{\theta} = V$, we obtain

$$N = \frac{\rho}{2} L^3 d \cdot V \cdot C_N$$

Similarly nondimensionalizing the turning moment caused by steering, we get

$$\frac{M\delta}{\frac{\rho}{2}\ L^2 d V^2} = C_M \cdot \delta$$

where C_M is the nondimensional turning moment coefficient. Then $M = \frac{\rho}{2}\ L^2 d V^2 \cdot C_M$. Consequently, nondimensional representation of K is obtained as follows:

$$K = \frac{M}{N} = \left(\frac{V}{L}\right) \frac{C_M}{C_N},$$

denoting $\frac{C_M}{C_N}$ by K',

$$K' = \left(\frac{L}{V}\right) K.$$

Next, nondimensionalizing I_e by dividing $\frac{\rho}{2}\ L^4 d$ considering I_e has a dimension of a moment of inertia, we get

$$I_e = \frac{\rho}{2}\ L^4 d\ C_I$$

where C_I is the nondimensional inertia of a ship.
Thus we obtain nondimensional T as follows,

$$T = \frac{I_e}{N} = \left(\frac{L}{V}\right) \frac{C_I}{C_N},$$

denoting $\frac{C_I}{C_N}$ by T',

$$T' = \left(\frac{V}{L}\right) T.$$

3.2 K-T DIAGRAM – CORRELATION BETWEEN INDICES K AND T

Surveying the physical constitution of the indices K and T, we find immediately that these two indices have a common denominator and therefore they are in a proportional relation. That is,

$$K' = \frac{C_M}{C_N} \quad \text{and} \quad T' = \frac{C_I}{C_N},$$

therefore

$$K' = \frac{C_M}{C_I} \cdot T'.$$

Considering that a normal force per unit area of a rudder is almost invariable for all usual ships with a similar stern arrangement, the numerator C_M (that is, the nondimensional turning moment coefficient) may be proportional to a relative rudder size $A_R/L\cdot d$ with an almost invariable proportionality constant, where A_R represents a rudder area. On the other hand, C_I may be nearly proportional to ∇/L^2d where ∇ represents a displacement volume, because C_I has been introduced through dividing an inertia of ship by $\frac{\rho}{2}\ L^4d$, and because a ratio of a radius of gyration to a ship length is nearly invariable for all usual ships, say, about 0.25. Then we get

$$K' = \begin{pmatrix}\text{almost invariable constant}\\ \text{for all usual ships}\end{pmatrix} \times \frac{A_R}{L\cdot d} \times \frac{L^2 d}{\nabla} \times T' \qquad [6]$$

Figure 12 illustrates this relation for K' and T' of approximately sixty merchant ships (refer to Table 2), which have been obtained through Kempf's standard maneuver test, while in the figure a reciprocal expression is taken for convenience of illustration. Namely, plotting

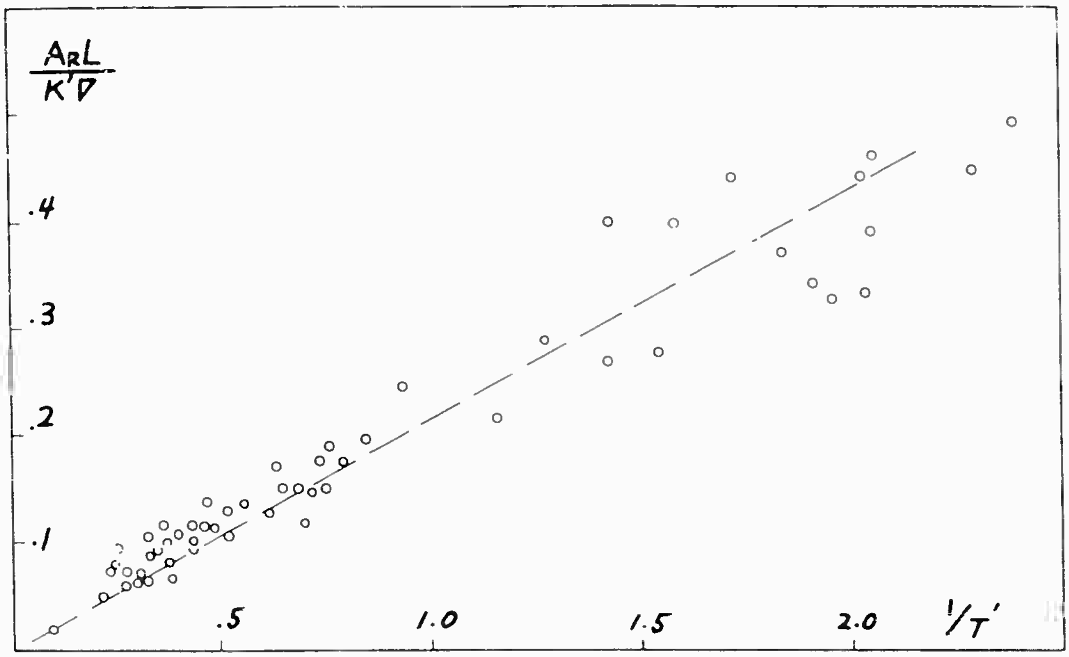

Figure 12 – K-T Diagram (Actual Ships)

$\frac{A_R L}{K'\nabla}$ against $\frac{1}{T'}$, plotted points gather along a straight line passing through the origin, the gradient of which indicates the foregoing "almost invariable constant" but reciprocally. Taking account of detailed differences in hull forms and rudder constructions among these ships and also of degrees of reliabilities of tests at sea which may be sometimes spoiled

because of weather conditions and personal errors, it may be considered that this result assures the validity of the foregoing reasoning on the whole.

Another similar result for a self-propelled, free-running model of a supertanker with alternative rudder sizes and various helm angles is shown in Figure 13. The result naturally is more satisfactory because a single hull form was used and test conditions were much better. The gradient of the straight line in Figure 13 equals that of Figure 12. The frequency response test described in Figure 13 is a somewhat different test procedure from Kempf's standard maneuver, and the indices obtained through it correspond to the ones for very small helm angle. (Refer to Reference 2.)

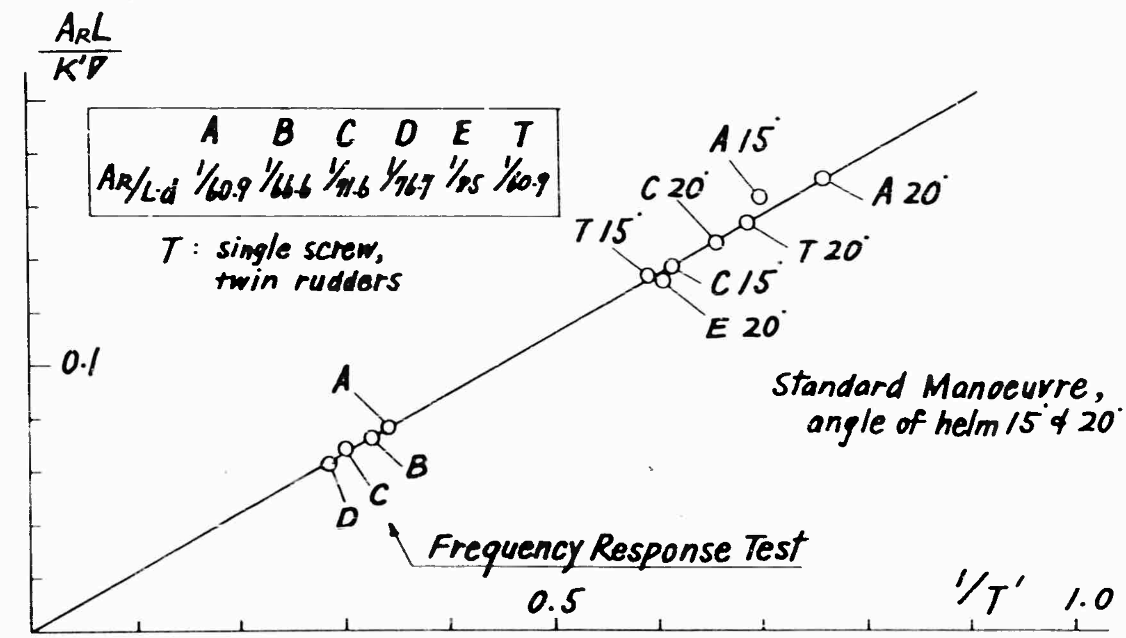

Figure 13 – K-T Diagram (a Model of a Supertanker)

The fact that K' and T' of a ship are connected to each other through the relation [6] provides important conclusions as follows:

1. The turning ability and quick response in steering or stability on course, which are the two essential abilities of a ship in steering, are not independent of each other but are bounded largely through the relative rudder size.

2. In consequence, if the relative rudder size is kept constant, an improvement of turning ability (increase of K) injures quick response in steering and stability on course (increase of T), and vice versa.

3. When one of the indices K' and T' is known, the other one may be estimated through this relation.

Referring to Figure 13, the results for different helm angles used in the tests correspond to different points but along the straight line. This means that the indices K and T vary with helm angles used so as to keep the proportional relation between them invariable. These circumstances are also found in the results for actual ships illustrated in Figure 12. The variation of K and T with helm angles used is considered to result from the so-called nonlinear effect in steering quality, which means a variation of coefficients in linear equations of steering motion depending largely on the intensity of motion. The effect for normal ships is fairly great so that the steering quality indices may be better defined by specifying a degree of intensity of motion, say, for a hard-over steering, a moderate one, or a course-keeping motion.

The variation of the indices, as pointed out previously, indicates, concerning the nonlinear effect, that:

1. The larger the helm angle used, the more intense is the ship motion. The smaller the index K, the smaller is the turning rate produced by a given helm angle. The smaller the index T, the more stable is a ship on course and also the quicker is her response in steering.

2. Since the variation of the indices is the same for each so as to keep the proportional relation between them, the nonlinear effect results largely from a variation of C_N; namely, that of a hydrodynamic damping coefficient against turning motion.

The first conclusion agrees with experiments, and the second seems to have a meaning worthy of notice concerning the nonlinear effect in steering motion.

Figures 12 and 13 remind us of a figure presented by Davidson and Schiff (viz, Figure 3 of Reference 4) which illustrates $\left(\frac{D}{L}\right)_{\min}$ × (relative rudder size) against their stability index p_1. Then D represents the turning diameter, and L the ship length. The value $\left(\frac{D}{L}\right)_{\min}$ is proportional to $\frac{1}{K'}$ considering that the maximum angle of helm is constantly 35 deg for all ships, and p_1 is nearly equal to $-\frac{1}{T'}$, as shown in Section 1.3. In addition, differences among ∇/L^2d of usual ships are fairly small. Thus the figure of Davidson and Schiff and the K-T diagram are essentially similar expressions. In the former expression, however, stability on course at straight running and turning ability at the hardest helm are related. This is the reason Davidson's plotting line does not pass the origin. Considering remarkable nonlinear effect for steering quality, it seems more reasonable to relate stability on course (or quick response) and turning ability under the same condition, as is done in the K-T diagram.

3.3 TURNING RESISTANCE AS A FUNCTION OF RUDDER AREA AND HULL SLENDERNESS FACTOR – ESTIMATION OF THE STEERING QUALITY INDICES K AND T

The foregoing discussion shows that $K' = \frac{C_M}{C_N}$ and $C_M \sim \frac{A_R}{L \cdot d}$ with a nearly invariable constant. Then we get $\frac{A_R}{K' \cdot Ld} \sim C_N$.

Namely, $\frac{A_R}{K' \cdot Ld}$ may be proportional to the nondimensional coefficient C_N of a hydrodynamic damping moment to resist turning motion, that may be called simply "turning resistance," with a nearly invariable proportionality constant over usual ships with a similar stern arrangement.

Figure 14 illustrates $\frac{A_R}{K' \cdot Ld}$ against $\frac{A_R}{L \cdot d}$ for a number of merchant ships full or half-loaded in Table 2. The figure indicates that turning resistance of a ship is essentially determined by her relative rudder size and is affected to some extent by a slenderness factor of a hull ∇/L^2d. Although other particulars, for instance, trim, C_b, B/d, and so on, may of course affect the turning resistance, the effect of the slenderness factor seems to be dominant so far as the data for full or half-loaded merchant ships are concerned.

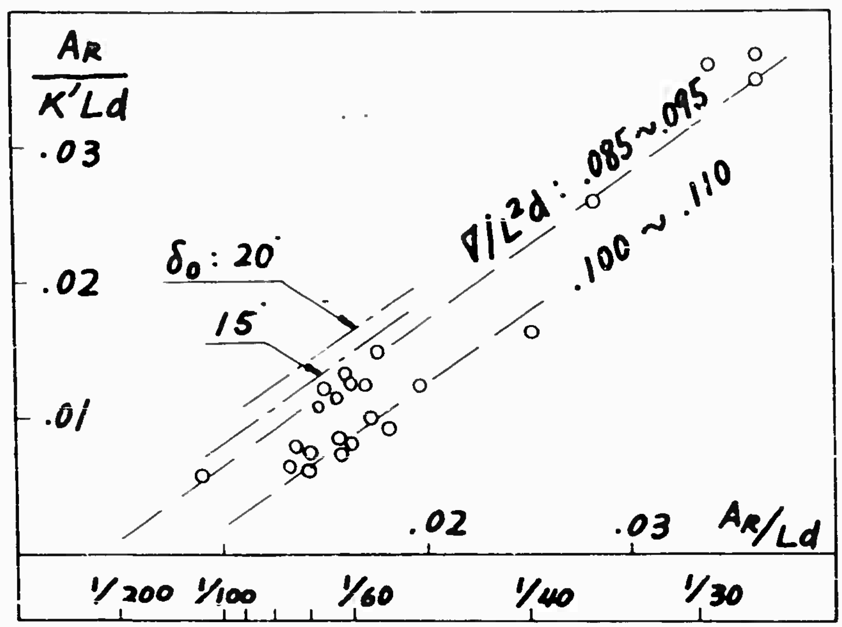

Figure 14 – Turning Resistance as a Function of Relative Rudder Size and Hull Slenderness

According to the figure, a ship has a generally negative turning resistance and consequently is unstable on course by her hull alone but is stabilized by adding a rudder of the proper size. Here the question arises: Why does a rudder contribute to turning resistance? The answer is as follows. As soon as a ship enters into a turn, drifting motion appears and accordingly the angle of incidence of current to a rudder decreases, which in turn yields a reduction of lateral force and turning moment caused by the rudder. This reduction of turning

moment may be considered as a resisting moment against turning. This damping effect caused by a rudder with ship motion may be understood more naturally by assuming that a ship with her rudder amidship is disturbed from a straight running. Circumstances are quite similar, however, even in cases where a certain helm angle is present. In other words, a normal force acting upon a rudder may be considered to be composed of two parts: one is proportional to a helm angle and in no relation to ship motion, and the other is proportional to a drift angle at rudder position (consequently nearly to turning angular velocity) and in no relation to a helm angle. All linear treatments on steering motion have been originally based upon this concept, whether or not specifying so. The former part is represented by terms proportional to a helm angle, and the latter is usually included in terms describing hydrodynamic forces acting upon a hull.

At any rate, it should be carefully noted that utility of a rudder is not merely to produce a turning moment but also to provide a damping against turning motion, which is so considerable that most ships may be almost unstable on course without rudders. Then obviously quick response in steering is also provided by a proper size of rudder, since this ability is originally the same one with stability on course, as is discussed in Section 1.3. In this connection, it is important to take account of the stabilizing effect of a rudder as well as of turning ability in selecting a rudder size; particularly in merchant ships, the stabilizing effect of the rudder may be the major consideration.

The plotted points of Figure 14 have been obtained from the standard maneuvers using a helm angle of 10 deg. Plotting also K's obtained from the maneuver tests with larger helm angles in the same form, the mean lines of the plotted points have a tendency to shift upwards and nearly in parallel, as is shown in Figure 14 by chain lines. This tendency indicates that turning resistance is not quite proportional to angular velocity, as is known as nonlinear effect in steering motion. (Refer to the last section.)

A practical use of the figure is to estimate roughly the index K' of a given ship knowing her relative rudder size and slenderness factor ∇/L^2d. Then taking the K' value, T' of the ship may also be estimated using the K-T diagram (Figure 12). This is the proposed procedure of predicting the maneuverability of a given ship. Accumulating much more data of this kind, the procedure may be expected to become more reliable and more widely practicable.

CONCLUSIONS

1. Referring to analyses of Kempf's standard maneuver tests for various types of ships, the steering motion of ships may well be described by the first-order equation of motion

$$T \frac{d\dot{\theta}}{dt} + \dot{\theta} = K\delta \qquad [2]$$

as has been predicted theoretically in Reference 2.

2. Maneuverability of a ship may be described essentially by the index of turning ability K and the index of quick response in steering and also of stability on course T.

3. Given the indices K and T, the motion of a ship in response to a given steering may be predicted with fair reliability.

4. A procedure of analyzing Kempf's standard maneuver test has been devised, and a calculating table for the purpose has been presented. The procedure has been practiced for results of standard maneuvers for approximately seventy ships, and the obtained steering quality indices K and T have been illustrated in Table 2. Examination of the data, sometimes considering also experimental results for self-propelled, free-running models, yields the following conclusions.

5. The indices K and T are in a proportional relation as follows:

$$K' = \begin{pmatrix} \text{almost invariable constant} \\ \text{for all usual ships} \end{pmatrix} \times \frac{A_R}{L \cdot d} \times \frac{L^2 d}{\nabla} \times T' \qquad [6]$$

This relation indicates that an improvement of turning ability injures quick response and stability on course, and vice versa, so far as a relative rudder size is kept constant.

6. The more intense the ship motion is, the smaller is the turning rate producible by a given helm angle and the more stable is a ship on course and also the quicker is her response in steering. This phenomenon, known as nonlinear effect in steering motion, results largely from a variation of a hydrodynamic damping coefficient against turning motion.

7. The hydrodynamic damping depends largely upon relative rudder sizes and to some extent upon hull slenderness factors ∇/L^2d. Most ships are almost unstable on course by her hull alone and are stabilized through adding a proper size of rudder. Then obviously quick response in steering is also provided by a proper size of rudder, since this ability is originally the same with stability on course, as has been discussed in Section 1.3. In this connection, it is important to take account of the stabilizing effect of a rudder as well as the turning ability in selecting a rudder size.

8. A procedure of estimating roughly the steering quality indices for a given ship is first to find K' from Figure 14 knowing the relative rudder size and slenderness factor ∇/L^2d, and then to find T' from Figure 12 (13) using the previously determined K'.

ACKNOWLEDGMENT

Acknowledgment is made of the cooperation by the staff of the Osaka University Tank.

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CONTROL OF ROLL DAMPING SYSTEM

by

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SUMMARY

Here are described some of the considerations involved in deciding upon the optimum combination of control signals for use in conjunction with the roll damping servomechanism involved when using activated fins.

The usual sensing device consists in a gyroscope sensitive to rolling velocity, but it can be shown both mathematically and by means of an electronic analogue that there is merit in using signals proportional to rolling acceleration and/or roll angle.

By means of the analogue technique, the effect of limiting fin capacity is shown, as well as the effect of lag in response.

It is shown that the use of phase advance can improve performance where lag exists. This result can also be achieved around the natural rolling frequency by inclusion of the acceleration term.

An approach to reproduction of the random sea is obtained by superimposing a complex consisting of a number of sinusoids of varying frequencies.

In this report the effect of feedahead is not discussed though it is considered worthwhile to study this possibility. A further valuable feature could consist in an arrangement whereby the loop ratio of the various signals could be varied over the working range of frequencies, so that an optimum setting can be maintained in all working conditions.

* *

As may be readily understood, the control of a roll damping system in such a manner as to produce optimum results in roll reduction represents an engineering problem of outstanding importance.

Most systems consist in what is usually termed *feedback* control, where a rate gyro or other sensing device signals a difference from the desired velocity (which will be zero) to the servomechanism. This can be effected in practice in various ways which will be described later, but essentially it is necessary for the ship to move in the rolling plane before any corrective moment can be applied by means of the fins.

Another and perhaps more effective control could be effected by what is termed *feedahead*. Here the disturbing moment is detected before the ship has responded and if *feedahead* is found practicable it has much to recommend it.

Considered in its elements, the problem consists in knowing the instantaneous magnitude of the disturbing moment applied or likely to be applied to the ship in the course of its passage through or across a wave system.

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[1]References are listed on page 321.

With this knowledge, it would, in theory at least, be possible to provide by some means an equal and opposite moment applied to the ship, which would result in a complete elimination of rolling. In fact, the roll would be killed before it could start.

This ideal situation is, however, difficult, if not impossible, to achieve in practice. Supposedly it would be possible, though rather difficult, to arrive at the heeling moment applied by the waves to the ship as the result of an integration of the pressure over the whole underwater surface of the ship at both sides.

The difference in hydrostatic head and therefore waterline level between one side of the ship and the other will give an indication of instantaneous wave slope relative to the ship.*

This could be formed by fitting pressure transducers at appropriate points on each side of the hull. A measurement of the pressure difference between one side and the other should provide information as to the immersion and therefore effective wave slope. If the wave slope relative to the horizontal is Φ ship's vertical axis relative the true vertical is θ then the angle measured from the pressure differences will be $\Phi \pm \theta$.

To find Φ therefore θ should be added or subtracted as appropriate.

Up to date the most practical means of applying a restoring moment resulting from a feedback signal has been found to consist in the technique of rotating fins or hydrofoils in such a manner that a lift force is developed due to the angle of incidence with the local flow. This lift force can be used as a restoring moment.

To a very reasonable approximation, the lift force for given flow velocity will be proportional to angle of incidence within the stall-free range. If the fin is placed at an angle of attack proportionate to velocity of roll, which is also conveniently enough found to be proportional to the angle of precession of a gyroscope whose axis lies in the athwartship plane, we shall go quite a long way toward reducing the roll. The amount of residual roll will, of course, depend upon the power of the fins as well as the *lag* in fulfilling the commands of the rate gyro.

In practice there is a limit to the size of the fins so that our problem consists in making the best use of the fin sizes available and practicable.

Although this is quite a neat solution to the feedback control problem, the rate gyro cannot in fact produce an instantaneous moment in the restoring direction unless we make the assumption that infinite fin power is available and that *lag* is nonexistent. Knowing out practical limitations in the matter of fin size and the inevitable delay, however slight, in fulfilling orders of the control signal, we must consider the extent to which a rate gyro will accomplish its job of roll reduction:

We can write the equation for ship rolling in a seaway as follows:

$$\Phi\,\Delta GM + D\dot{\Phi} = J\ddot{\theta} + D\dot{\theta} + \Delta GM\theta$$

when Φ = wave slope of sea,

θ = roll angle of ship,

---

*It has to be realized that the pressures sensed by transducers at the ship's side will be affected by the shape of the ship's wave, also by transient and random velocity heads due to orbit velocities in waves.

J = the inertia,

Δ = the displacement of the ship,

GM = distance between metacenter and center of gravity of ship, and

D = the natural viscous damping of the vessel.

Making various simplifying assumptions which include sinusoidal motion of disturbing moment due to waves, we can obtain the ratio of roll angle to wave slope for the case of the ideal rate control

$$\frac{\theta}{\Phi} = \frac{1 + j2\,\psi_N\,x}{1 - x^2 + j2x\,(\psi_N + \psi_F)}$$

where ψ_N = ratio of natural damping to critical damping,

and ψ_F = ratio of rate controlled fin moment to critical damping moment, and

x = ratio of frequency of wave encounter to natural rolling frequency.

If we add an acceleration term to the signal, this equation becomes:

$$\frac{\theta}{\Phi} = \frac{1 + j2\,\psi_N\,x}{1 - (1 + A)\,x^2 + j2\,x\,(\psi_N + \psi_F)}$$

Values of $\frac{\theta}{\Phi}$ are plotted for a range of values of x (see diagram) for three conditions where (1) no roll damping fins are operative; (2) where roll damping fins of ideal characteristics are operated by signals from a rate gyro (i.e., the fins are moved proportionately to rolling velocity); and (3) where fins are operated by signals proportionate to both velocity and acceleration.

It will be observed from Figure 1 that for this somewhat hypothetical case the addition of the acceleration term increases the residual roll, i.e., reduces effectiveness in comparison with the velocity term only for frequency of encounter less than that of the natural rolling frequency, while improved performance can be anticipated for higher frequencies of encounter.

In other words, the acceleration term is only of value in this idealized case for the higher frequency cases where short periods of wave encounter a ship of greater natural rolling period.

For the lower frequencies of encounter, an equivalent effect in reducing the residual roll can be achieved by a signal proportional to roll angle from a positional gyro or pendulum.

It must immediately be said that this case is not by any means of necessity the state of affairs met with in the random sea of nature.

The amount to which the realistic sea input departs from the sinusoidal model is a matter of some conjecture and will probably require the reduction of much statistical data before anything approaching a precise answer can be given.

However, it should be mentioned that Mr. John Beli, who has been mainly responsible for the design of the Muirhead gyros we use, is clear that the use of *compensated* control

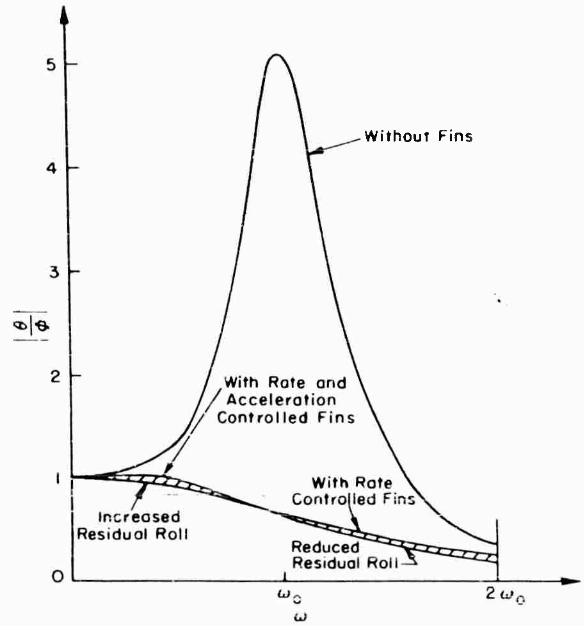


Figure 1

offers substantial advantage in this respect. Mr. Bell claims that the classic sinusoidal assumption is replaced in practice by an aperiodic motion so that the acceleration term will prove highly effective in putting the fins into the correct position to generate a stabilizing moment at an earlier moment than would be the case where a rate term only was used.

There are various methods which can be used to insert an acceleration term in the control signal. The easiest and one which is quite frequently used in other servocontrol systems such as guided missiles, etc. is to differentiate electronically from the rate signal, thereby obtaining a value for acceleration.

The method used by Muirhead is to differentiate by means of mechanical linkage; alternately it is not, of course, impossible to provide an angular accelerometer.

In an effort to arrive at an idea qualitatively as to how much advantage in terms of roll reduction we can anticipate by adding to the control signal terms proportional to acceleration and/or amplitude of roll, Vosper has investigated the matter on an Electronic Analogue.

MATHEMATICAL REPRESENTATION OF THE ROLLING SHIP IN A WAVE INFLUENCED BY A ROLL DAMPING SYSTEM

The ship rolling in a seaway was assumed to be a body with inertia, natural stability, and viscous damping relative to the forcing function. Thus basically, the equation of the vessel is of the form:

$$\text{Forcing function} = J\ddot{\theta} + \Delta GM\theta + D\,(\dot{\theta} - \dot{\Phi})$$

In the case of a ship rolling, the forcing function is due to the wave slope of the sea and so the equation becomes:

$$\Phi\Delta GM - \dot{\Phi}D = J\ddot{\theta} + \Delta GM\,\theta + D\dot{\theta} \quad \ldots \quad \ldots \quad \ldots \qquad [1]$$

(bearing in mind that for S.H.M. $\frac{\Delta GM}{J} = \omega_0^2$)

This equation may be normalized by dividing all through by J to become:

$$\Phi\,\omega_0^2 + \dot{\Phi}\,2\,\psi_N\,\omega_0 = \ddot{\theta} + 2\,\psi_N\,\omega_0\,\dot{\theta} + \omega_0^2\,\theta \qquad [2]$$

The basic arrangement in the analogue simulates this equation.

The addition of idealized roll damping fins controlled by a rate signal is simulated by differentiating the output of this system and feeding it back with the appropriate phase and amplitude to the forcing function input. Thus we now have an analogue solving:

$$\Phi\,\omega_0^2 + \dot{\Phi}\,2\,\psi_N\,\omega_0 - 2\,\psi_F\,\omega_0\,\dot{\theta} = \ddot{\theta} + 2\,\psi_N\,\omega_0\,\dot{\theta} + \omega_0^2\theta \qquad [3]$$

This is the arrangement shown in Figure 2.

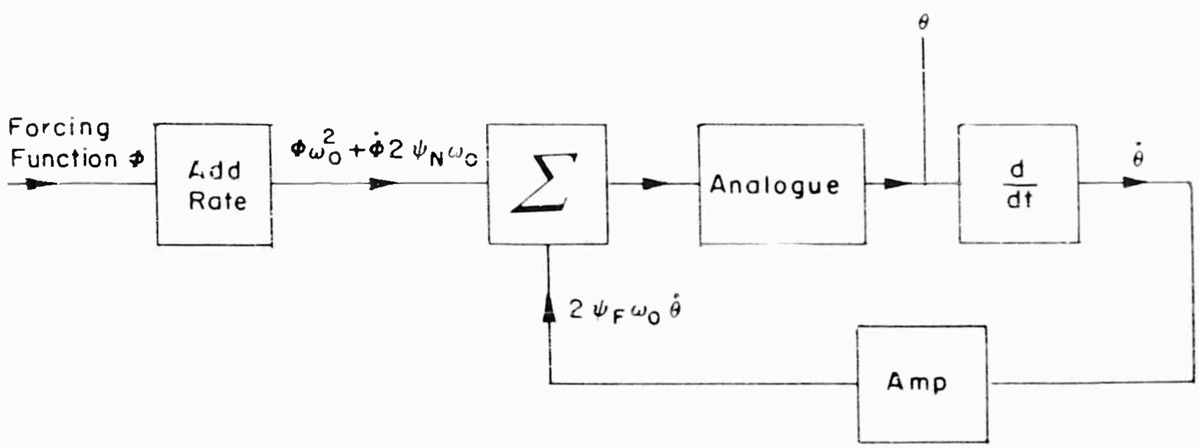


Figure 2

Where in practice ψ_F would be given by:

$2\,\psi_F\,\omega_0\,J$ = Fin moment per radian per second of roll assuming that fin lift is proportional to angle of incidence.

If the control of the fin moment is made more complex by the addition of position and acceleration control terms, then Equation [3] becomes:

$$\Phi\,\omega_0^2 + \dot{\Phi}\,2\,\psi_N\,\omega_0 = \ddot{\theta} + 2\,\psi_N\,\omega_0\,\dot{\theta} + \omega_0^2\,\theta + 2\,\psi_F\,\dot{\theta}\,\omega_0 + A\ddot{\theta} + P\omega_0^2\,\theta \qquad [4]$$

Where A and P are nondimensional terms given by:

$$A\dot{J} = \text{Fin moment per radian per sec per sec of roll.} \qquad [5]$$

$$P\omega_0^2 J = \text{Fin moment per radian of roll.} \qquad [6]$$

This arrangement is simulated by adding to the $\dot{\theta}$ signal, itself differentiated and a fraction of θ, as shown in Figure 3.

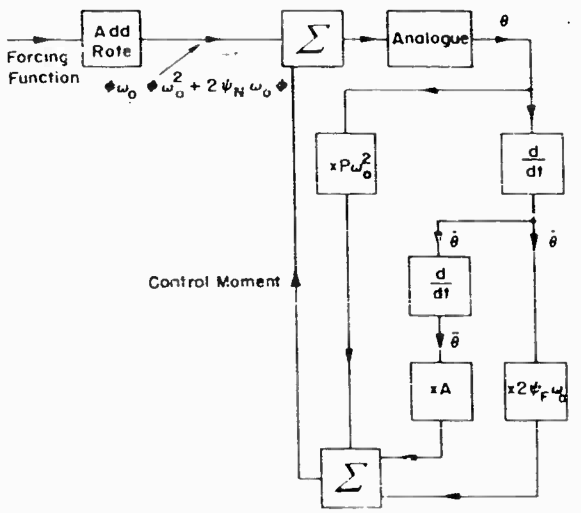


Figure 3

In practice, a lag occurs between the production of the control signal and the application of the fin moment, and this is included by inserting various first-order lags between the combined fin moment and its point of application in the system. This is shown in Figure 4 whose corresponding equation is:

$$\Phi\,\omega_0^2 + \dot{\Phi}\,2\,\psi_N\,\omega_0 = \ddot{\theta} + 2\,\psi_N\,\omega_0\,\dot{\theta} + \omega_0^2\,\theta + e^{-j\omega}\;(2\,\psi_F\,\omega_0\,\dot{\theta} + A\ddot{\theta} + P\omega_0^2\,\theta) \qquad [7]$$

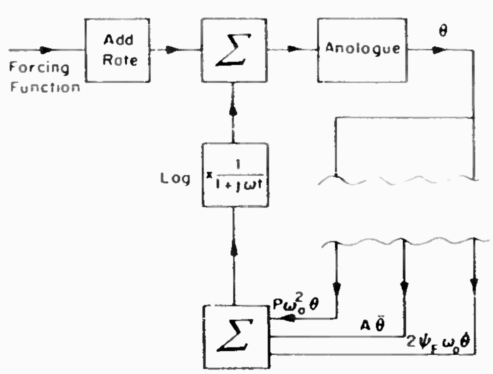
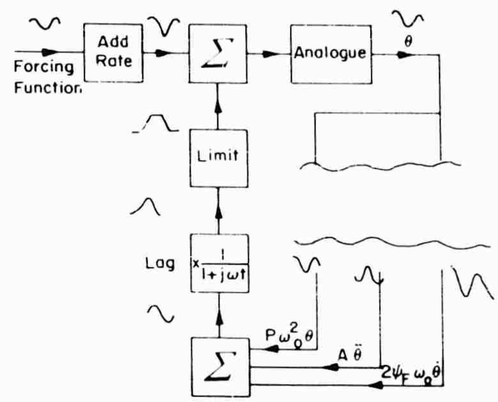


Figure 4

A further inclusion in the analogue which cannot be expressed simply by a mathematical expression is a limiter which prevents the control moment rising above a certain value.

This represents the effect of a limit to the maximum fin moment which can be applied in practice due to stalling. The complete analogue is thus as shown in Figure 5.

Figure 5



Some predictions of the results to be expected can be made by examination of the relevant equations for a steady-single-frequency sinusoidal disturbance. For this condition Equation [3] becomes:

$$\Phi\,(\omega_0^2 + j2\,\psi_N\,\omega\omega_0) = (-\omega^2 + j2\,\psi_N\,\omega\omega_0 + \omega_0^2)\,\theta + j2\,\psi_F\,\omega\omega_0\theta$$

which on division by ω_0^2 becomes:

$$\Phi\,(1 + j2\,\psi_N\,x) = (1 - x^2 + j2\,\psi_N\,x + j2\,\psi_F\,x)\,\theta \qquad [8]$$

This clearly shows that the fins should be most effective at resonance, and the ratio of the roll with fins to the roll without fins is shown plotted in Figure 6 for various frequencies. This also may be deduced by consideration of the phase of a pure rate signal at various frequencies. Figure 7 shows that above and below resonance the rate signal will not be opposing the forcing function as well as it does at resonance and so its effect may be expected to be reduced. Similar diagrams (Figure 7) for the position and acceleration vectors show that they will be most effective below and above resonance, respectively. These hypotheses are borne out by the results obtained on the simulator which are shown photographically in Figure 8.

NOTE: It will be observed that we have selected $\left|\frac{\theta}{\Phi}\right|$ as the criterion of system performance in roll damping which actually compares roll angle θ with wave slope. An alternative would be a consideration of the relationship between $|\theta|$ fins off and $|\theta|$ fins on. This can be arrived at by dividing Equation [2] by Equation [4].

Since the acceleration signal varies as the square of the frequency and the position signal is constant and as they are in antiphase, by arranging that these extra control signals

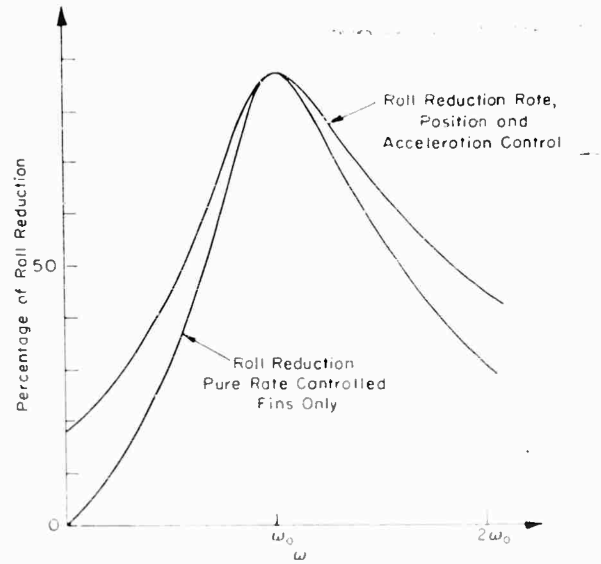


Figure 6

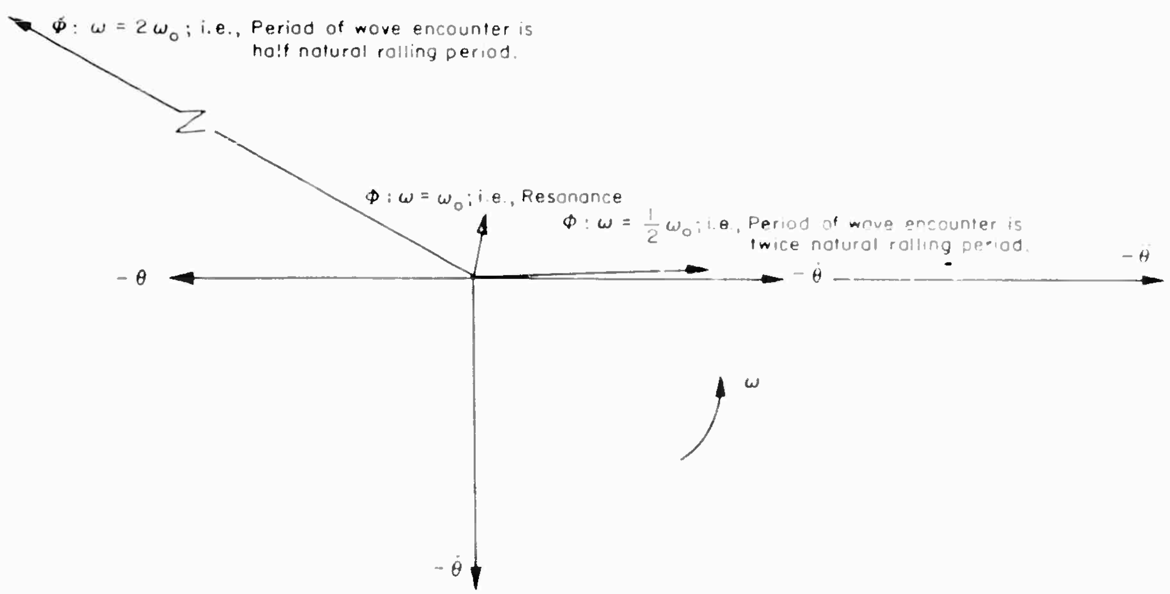
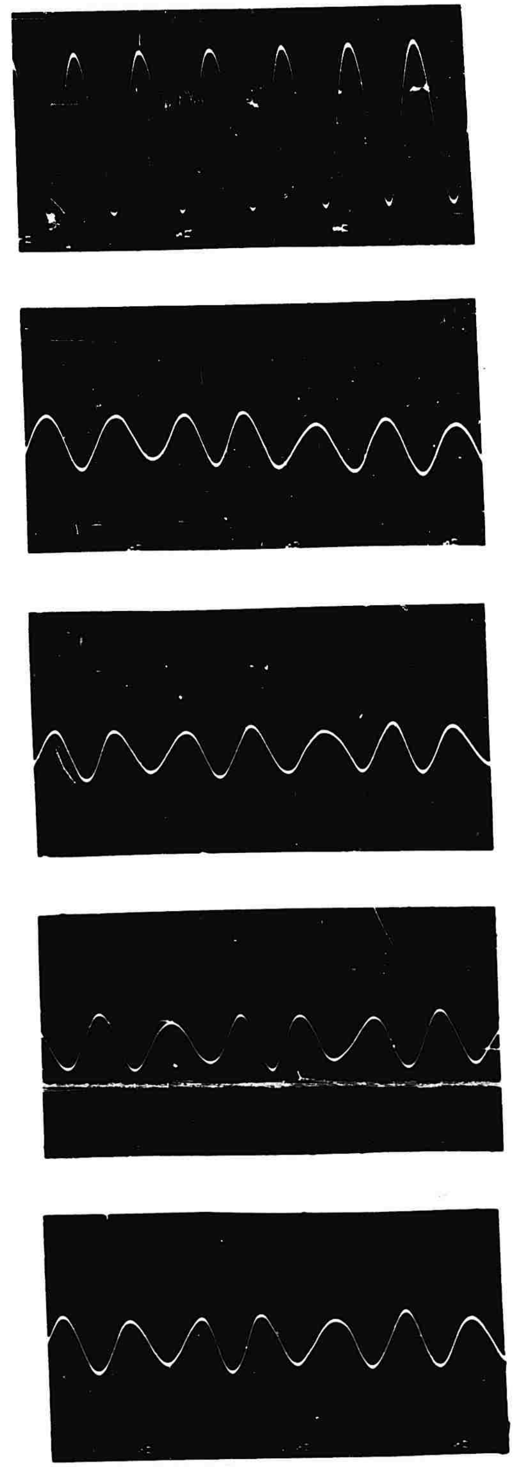


Figure 7



FINS OFF
Average amplitude 2.6 cm

$\dot{\theta}$ CONTROLLED FINS
Amplitude 0.9 cm

$\dot{\theta}$ and θ CONTROLLED FINS
Amplitude 0.85 cm

$\dot{\theta}$ and $\ddot{\theta}$ CONTROLLED FINS
Amplitude 0.95 cm

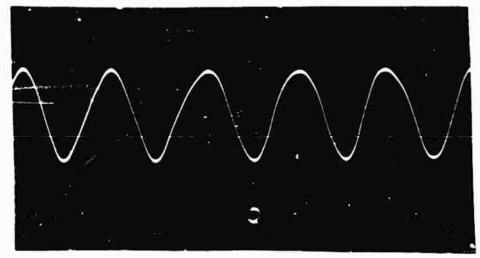
θ, $\dot{\theta}$, and $\ddot{\theta}$ CONTROLLED FINS
Amplitude 0.9 cm

Figure 8a – Recordings Taken at Resonance of Roll Angle

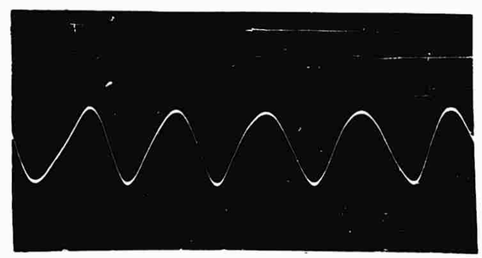
(Note: Distortion of sine waves is due to 50-cps Pickup.)

FINS OFF
Amplitude 2.75 cm

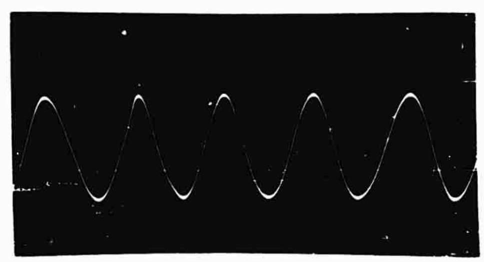
$\dot{\theta}$ CONTROLLED FINS
Amplitude 1.5 cm



$\dot{\theta}$ and θ CONTROLLED FINS
Amplitude 1.3 cm



$\dot{\theta}$ and $\ddot{\theta}$ CONTROLLED FINS
Amplitude 1.7 cm



$\dot{\theta}$, θ, and $\ddot{\theta}$ CONTROLLED FINS
Amplitude 1.4 cm

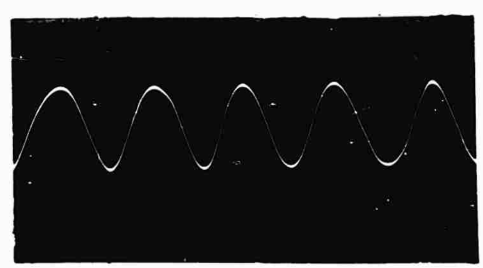
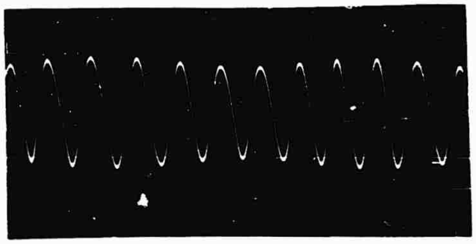


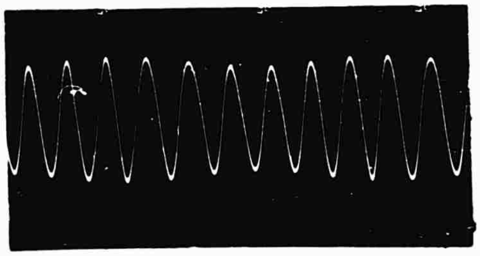
Figure 8b – Recordings of Roll Angle Taken Below Resonance

FINS OFF
Amplitude 2.65 cm

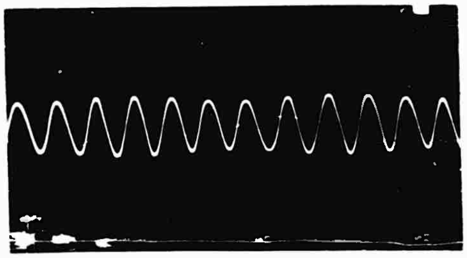
RATE-CONTROLLED
FINS ON
Amplitude 1.65 cm



$\dot{\theta}$ and θ CONTROLLED FINS
Amplitude 1.85 cm



$\dot{\theta}$ and $\ddot{\theta}$ CONTROLLED FINS
Amplitude 0.95 cm



$\dot{\theta}$, $\ddot{\theta}$, and θ CONTROLLED FINS
Amplitude 1.3 cm

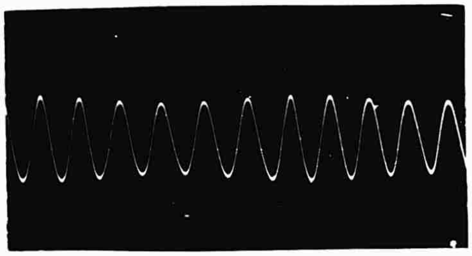


Figure 8c – Recordings of Roll Angles above Resonance

are equal and opposite at resonance, then automatically the position signal will be predominant below resonance and the acceleration predominant above, as shown in Figure 9. This was the condition arranged for the combined $\dot{\theta}$, θ, and $\ddot{\theta}$ control whose results are shown in Figure 8.

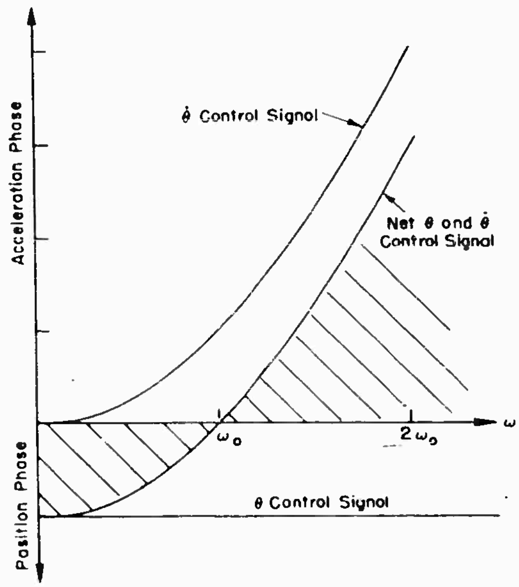


Figure 9 – Magnitude of Additional Control Moment for Constant Roll Angle

The effect of a lag at resonance is shown in Figure 10, and the vectors show that the effect is likely to be overcome by the addition of acceleration control, or phase advance applied to pure rate signal could have some effect. This was confirmed by using the analogue, the results being shown in Figure 11.

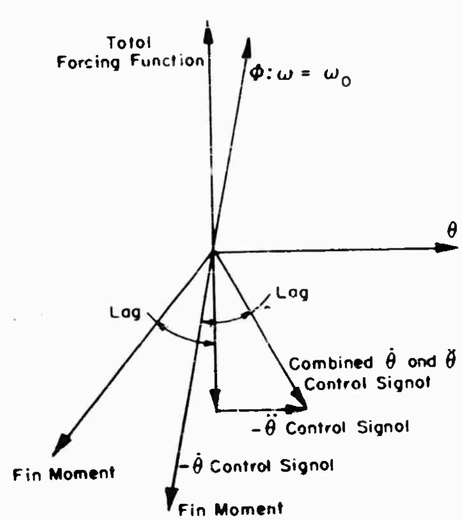
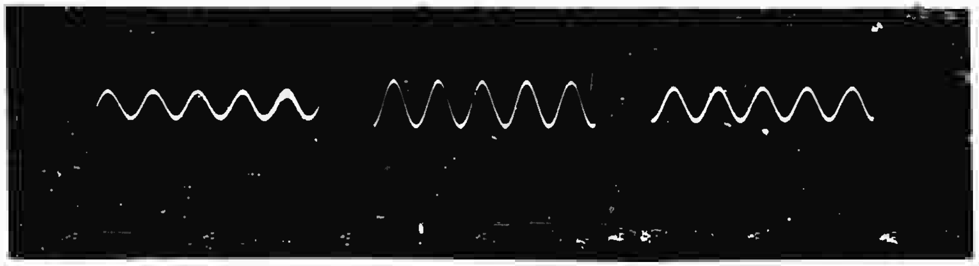


Figure 10 – The Total Forcing Function $\Phi\ (1 + j2\psi_N x)$ Acts at Right Angles to the Roll Amplitude



(a) (b) (c)

Figure 11 – Residual Roll at Synchronism

(a) – Pure rate control
(b) – With lag
(c) – Rate control with lag partly overcome by acceleration control

As indicated above, the sea is not a steady sinusoidal disturbance but contains many different frequencies. A first approximation to this is to use a mixture of sinusoidal forcing functions on the analogue and this has been done, up to six different frequencies being simultaneously applied. The results show that the trends anticipated from consideration of the forcing function composed of a single sinusoidal frequency are to a very close approximation applicable.

Figure 12 shows output from the rolling ship analogue when the disturbing sea is composed of a complex of three frequencies whose amplitudes are equal.

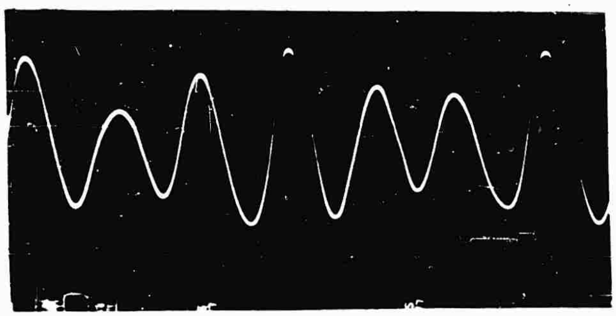
When rolling freely, as in (a), the tendency is for the frequency of the natural rolling period to predominate.

Figure 12b represents the residual roll from the same sea input when fins are controlled by a signal proportionate to rolling rate [$\dot{\theta}$].

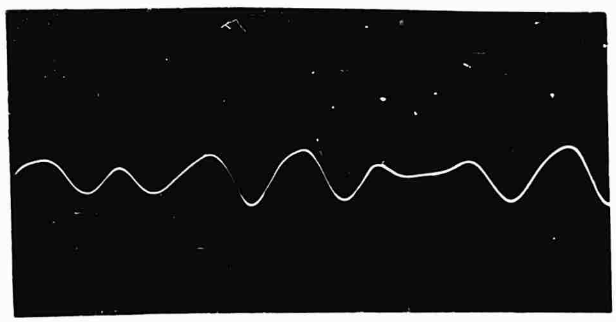
Figure 12c represents residual roll with the addition of terms proportional to acceleration and position [$\ddot{\theta}$ and θ] added to the rate signal. No very noticeable improvement results.

Figures 12d, e, f are recordings of the roll when the forcing function contains the same three frequencies. Their amplitudes were adjusted so that they individually caused the same amplitude of roll of the vessel rolling freely. Figures 12e and f show that in this case the addition of the acceleration and position terms to the control signal moments has a somewhat more noticeable effect on residual roll.

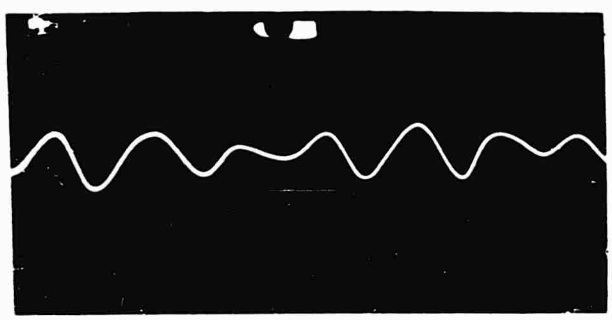
The closeness with which these results correspond with actual seagoing practice is a matter of some conjecture but they can be considered to be qualitative as far as the effect of the various control signals is concerned. Analysis of wave patterns suggests that sea waves are akin to white noise in that they contain all frequencies. However, seamen have long been acquainted with the apparently varying characteristics of seas in different parts of the world, which may indicate a predominance of certain frequencies in certain areas. With this it must



(a) ROLL ANGLE, NO FINS



(b) RESIDUAL ROLL ANGLE
Pure rate-controlled fins

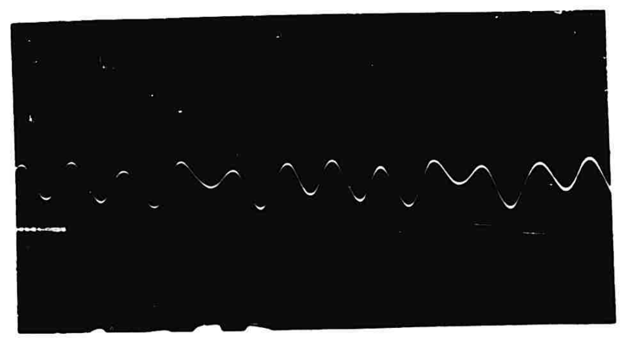


(c) RESIDUAL ROLL ANGLE
Fins controlled by position,
Rate, and acceleration

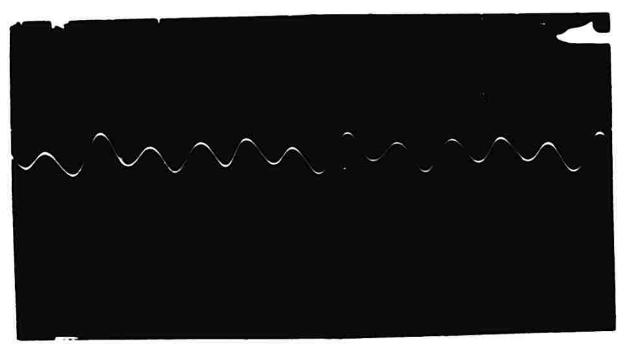
Figures 12a, b, c – Roll Angles Due to Forcing Function Comprising Equal Amplitudes of a Complex Sea of Three Frequencies to Represent the Random Sea

(d) ROLL WITH FINS OFF

(e) RESIDUAL ROLL WITH PURELY
RATE-CONTROLLED
FINS OPERATING



(f) RESIDUAL ROLL WITH CONTROL
MOMENTS DUE TO POSITION,
RATE, AND ACCELERATION



Figures 12d, e, f – Rolling Due to Forcing Function Comprising the Three Frequencies Previously Used, Their Amplitudes Adjusted to Make Equal Contribution to the Freely Rolling, Roll Amplitude

be borne in mind that the dimensions of a vessel, compared with the wave length, will be significant. Wave length is a function of wave velocity and frequency. The effects of frequency have been investigated above. Apparent wave velocity will depend upon the absolute wave direction and velocity in an area and the direction and speed of the ship. This can only be assessed by a very careful statistical analysis of the results of many trials, with wave recordings being made of the area in which the vessels are being run.

ACKNOWLEDGMENT

Sincere acknowledgment is made to Mr. John Bell, M.Sc., of Muirhead & Co. who have supplied all the control equipment used up to date in connection with the Vosper roll damping systems.

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*Senior Research Engineer, Feedback Controls Inc.

TURNING CHARACTERISTIC COEFFICIENTS FOR A CARGO SHIP AND A DESTROYER

EFFECT OF A SKEG
EFFECT OF A CHANGE OF PROPORTIONS

by

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(Paris Model Basin)

ABSTRACT

The turning qualities of a ship are characterized by the forces developed when three principal parameters are varied: turning radius, drift angle, and rudder angle.

The determination of these forces is the subject of systematic tests at the Paris Model Basin. The results for a model of a cargo ship and for a model of a destroyer are given; the effect of a skeg (rudder amidship) and the effect of a change of proportions are discussed.

I. SETTING THE PROBLEM

1. INTRODUCTION

Ship turning is the whole which results from the individual qualities of the hull and the rudder, from their reciprocal action, and from the effect of the screws. Therefore, to make progress in understanding turning, it is not sufficient to observe the natural motions of the ship; it is also necessary to know all the forces involved.

A theoretical approach is possible. The ship is likened to a wing, but the span (vertical) of this wing is very small and the approximations of classic aerodynamics are not valid, particularly to account for the existence of a lift when the hull is in a turn with a zero drift angle. By means of new methods,[1, 2] it is possible, however, to calculate the forces acting on a turning ship and also to explain results already known experimentally (for example, those concerning the course stability) and even to predict new ones. But these methods require complicated calculations, particularly for full bodies; it is then useful to derive from experiments the turning data.

Davidson and Schiff[3] have opened the way; they have determined the stability of straight-line motion of a ship in analyzing, for the conditions of free turning, the distribution of forces between the hull and the rudder. It is possible to go further and to determine the forces acting on a model of which the turning radius, drift angle, and rudder angle are arbitrarily fixed.

A systematic study has been undertaken in this direction at the Paris Model Basin. The field of possible investigations being very extensive, most of the tests carried out up to now have concerned the hull itself; bare hull and hull with stern appendages (skeg and rudder amidship), the interactions between the hull and the deflected rudder, and the effect of screws, have been the subject only of individual studies.

[1]References are listed on page 338.

2. EXPERIMENTAL PROCEDURE

Forces are measured on a model in straight and curve courses. The straight-course tests are carried out in a rectilinear tank; the turning tests are carried out in a circular tank of 65-meter diameter provided with a rotating arm under which is fitted a carriage, the position of which is adjustable. Although the tank is relatively large, it is necessary, for this kind of study, to use relatively small models in order that the ratio L/R of the length of the model to the turning radius may have values close to zero; the normal length of the models is 2.5 meters but some smaller models have also been used. For comparison with the simple theoretical method, it is necessary to get rid of the effect of ship waves, hence to operate at reduced speed: 0.80 m/s generally. The effect of heel is avoided by displacing weights so that the model is kept upright. The length and the speed of the model being small, in spite of turbulence stimulation* a scale effect for predicting the full-scale ship is expected, but if this effect may be relatively important for the longitudinal force (which does not interest us here) we think that it is very small regarding the transverse force; hence, at least for comparison between different solutions, model results are applicable to ships.

The horizontal forces are measured with a balance; they are resolved by the balance into a longitudinal force and two transverse forces passing through fixed points, one ahead and one behind the center of gravity.

3. ANALYSIS OF TEST RESULTS

Symbols

Length of ship	L
Draft	T
Longitudinal area of bare hull	S_0
Projected area of the rudder	S_1
Projected area of the skeg	S_2
Total longitudinal area	S
Speed of the center of gravity	V
Turning radius	$R \left[\frac{R}{L} = k'\right]$ **
Drift angle	δ
Rudder angle	α
Transverse force	Y

*Pins are usually fitted 1/20 length from the bow and on the appendages.

**k' is the derivative of heading k with respect to the time if the time unit is the time which the ship takes to travel its own length.

Yawing moment (moment of the force related to G) $\quad N$

Transverse force coefficient* $\quad C_Y = \dfrac{Y}{\pi \dfrac{\rho}{2} SV^2 \dfrac{2T}{L}} = \dfrac{Y}{\pi\rho SV^2 \dfrac{T}{L}}$

Yawing moment coefficient* $\quad C_N = \dfrac{N}{\pi \dfrac{\rho}{2} SV^2 \dfrac{2T}{L} \times \dfrac{L}{2}} = \dfrac{N}{\dfrac{1}{2} \pi\rho STV^2}$

In the case of Figure 1, δ, k', α, Y, and N are all positive.

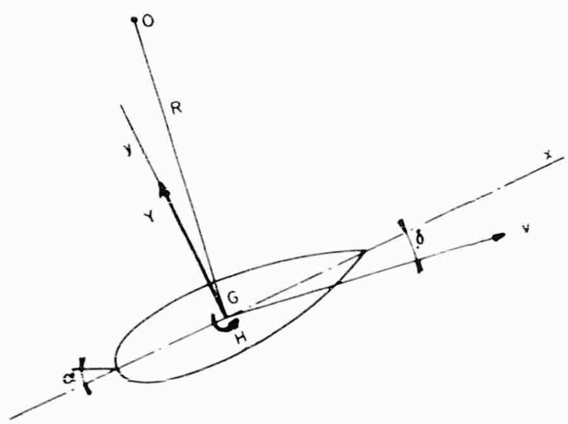


Figure 1

Characteristic Coefficients

In the absence of separation of the fluid, that is to say, with a reasonable angle of incidence, C_Y and C_N are uniform and odd functions of δ, $\frac{k'}{2}$, and α. For $\alpha = 0$, and limiting to terms of the third order (generally sufficient in practice) C_Y and C_N may be expressed in the following form:**

---

*Casal demonstrated that for a wing of very small aspect ratio, hydrodynamics coefficient C_{YH} and C_{NH} are expressed by:

$$C_{YH} = \delta + \frac{k'}{2}$$

$$C_{NH} = \delta - \frac{k'}{2} \qquad (\delta \text{ in radians.})$$

**It is possible to have second-order terms which will be, for example, in the form $\delta \times |\delta|$, but experience shows that sufficient adjustments may be obtained by assuming them zero.

$$C_Y = a_1\ \delta + b_1 \frac{k'}{2} + a_3\ \delta^3 + b_3\ \delta^2 \frac{k'}{2} + c_3\ \delta \left(\frac{k'}{2}\right)^2 + d_3 \left(\frac{k'}{2}\right)^3$$

$$C_N = a_1'\ \delta + b_1' \frac{k'}{2} + a_3'\ \delta^3 + b_3'\ \delta^2 \frac{k'}{2} + c_3'\ \delta \left(\frac{k'}{2}\right)^2 + d_3' \left(\frac{k'}{2}\right)^3$$

where δ is expressed in radians.

The turning of a body is therefore characterized by 12 coefficients; experience shows, however, that in most cases the number of coefficients may be reduced to 10 (d_3 and $d_3' = 0$), and even often to 8 (c_3, d_3, c_3' and $d_3' = 0$).

The coefficients of the first order have a particular importance because they characterize stability of route with zero rudder angle. The condition of stability, at first given in analytic form by Contensou,[4] has been expressed in geometric form by Dieudonné.[5] The ship being in a straight course ($k' = 0$), the transverse force has a limiting position T on the axis when δ tends to zero: $GT = \frac{a_1'}{a_1}$; likewise, the ship turning without drift angle ($\delta = 0$), the transverse force (hydrodynamic force and centrifugal force) has a limiting position D on the axis when k' tends to zero: $GD = \frac{b_1'}{b_1}$. It is shown that the straight-course condition is stable if D is outside the segment GT. In open water, T and D being practically ahead of G, the straight-course regime is stable if D is ahead of T.*

The coefficients of the third order are important for free turning with relatively large rudder angles; they may account for the fact that Ship A turns poorer than Ship B for small rudder angles and that, on the contrary, A turns better than B for large rudder angles.

When the rudder is not amidship, other coefficients are to be considered in respect to α ; here also their number may be often reduced, especially in the case where the rudder is well clear from the hull (hull-rudder interactions being then reduced).

---

*Then, $\frac{b_1'}{b_1} > \frac{a_1'}{a_1}$

II. CHARACTERISTIC COEFFICIENTS FOR A CARGO SHIP*

1. FORM AND PROPORTIONS

Tests were carried out on a 4200-ton d-w cargo vessel, the characteristics of which are given in Table 1.

TABLE 1

Length (W.L.)	$L = 114.50$ m	
Breadth	$B = 15.50$ m	$\frac{B}{L} = 0.135$
Draft	$T = 6.00$ m	$\frac{T}{L} = 0.0524$
Displacement	$W = 7.320$ m^3	$\frac{W}{LBT} = 0.687$
Longitudinal area without appendages	$S_0 = 647$ m^2	$\frac{S_0}{LT} = 0.942$
Projected area of the rudder	$S_1 = 10.64$ m^2	$\frac{S_1}{S_0} = 0.0165$
Distance from G to AP (W.L.)	$\xi = 54.45$ m	$\frac{\xi}{\frac{L}{2}} = 0.951$

The arrangement of the propeller aperture and of the rudder are shown in Figure 2.

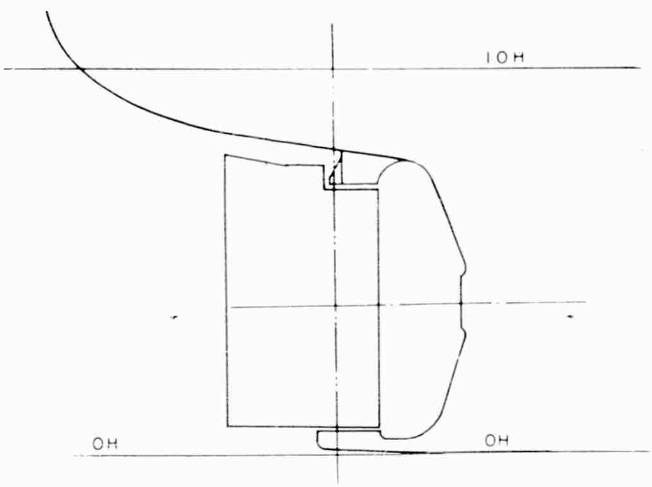


Figure 2

*Study developed for the French "Institut de Recherches de la Construction Navale."

2. TEST PROGRAM

Two forms have been tested: Form 1 is geometrically similar to the full-scale ship; Form 2 is obtained by expansion of Form 1, draftwise:*

$$\frac{T_2}{T_1} = \frac{1}{0.9} = 1.11$$

The model scales were respectively 1/66.67 and 1/74.07.**

The following tests have been carried out:

On the two forms, measurements of forces on the bare hull and on the hull provided with a rudder (amidship) for $-10° < \delta < 10°$ and $-0.3 < k' < +0.3$.

On Form 1 only, measurement of forces for $-5° < \alpha < +25°$

a. $\delta = 0,\ 0 < k' < 0.25$

b. $-2° < \delta < +8°,\ k' = 0.$

The test speed corresponded to 15 knots for the ship.

Moreover, free-turning tests have been made on large models (scales 1/20 and 1/22.22, respectively), fitted with a propeller.

3. CHARACTERISTIC COEFFICIENTS OF THE HULL

In the range of δ and k' studied, the variations of C_Y and C_N are linear functions of k'; the coefficients c_3, d_3, c_3' and d_3' are then zero.. The values of the eight remaining coefficients are given in Table 2.

The presence of the rudder increases b_1 and decreases a_3' and b_3'.

The increase in draft (passage from Form 1 to Form 2) increases b_1 total and decreases b_3 and b_3'.

4. STABILITY ON STRAIGHT COURSE

Table 3 indicates the limiting position T (towing with drift angle) and D (turning without drift angle).

Point T is always beyond segment GD; consequently the straight-course regime is unstable in all cases.

The presence of the rudder decreases the instability but relatively little because the rudder, placed behind the hull, is masked by it.†

The influence of draft on stability is not clear.

---

*Expansion of bare hull and rudder.

**The models used were thus smaller than normal.

†It is certain that the effect of the rudder would have been increased if the tests had been made with the propeller working.

TABLE 2

	Coefficients Relative to C_Y				Coefficients Relative to C_N			
	a_1	$b_1{}^*$	a_3	$b_3{}^*$	a_1'	b_1'	a_3'	b_3'
Form 1 without rudder	0.72	0.01 −1.11	10.5	8.5 9	0.83	−0.58	0	− 8.5
with rudder	0.71	0.23 −0.95	10	9.5 10	0.83	−0.62	−1.95	− 10
Form 2 without rudder	0.74	0.06 −1.02	10	6.5 7	0.82	−0.58	−0.40	− 10
with rudder	0.71	0.20 −0.87	11	6 6.5	0.78	−0.50	−1.20	−14.5

*First line: coefficient relative to C_Y hydrodynamic.
Second line: coefficient relative to C_Y total (hydrodynamic + centrifugal).

TABLE 3

	$\frac{GT}{L/2}$	$\frac{GD}{L/2}$	$\frac{TD}{L/2}$
Form 1 without rudder	1.16	0.52	−0.64
with rudder	1.17	0.66	−0.51
Form 2 without rudder	1.11	0.57	−0.54
with rudder	1.10	0.58	0.52

5. FORCES ON THE RUDDER

Measurements have been made only on Form 1, for $k' = 0$, δ variable and for $\delta = 0$, k' variable. Since the measurements were not made on rudder and hull separately, the efficiency of the rudder is the difference between values obtained when the rudder is deflected and when it is at zero:

$$\Delta C_Y = C_Y(\alpha) - C_Y(0) = \text{effective force on the rudder}$$

$$\Delta C_N = C_N(\alpha) - C_N(0) = \text{turning moment of the effective force on the rudder}$$

The results are as follows:

For $-5° < \alpha < +25°;\ -2° < \delta < +8°;\ k' = 0$

$$\left|\begin{array}{l} \Delta C_Y = -0.060\ \alpha\ -4\ \alpha\ \delta^2 + \alpha^2\ \delta \\ \Delta C_N = \ \ 0.053\ \alpha\ +2.5\ \alpha\ \delta^2 - 0.5\ \alpha^2 \delta \end{array}\right.$$

For $-5° < \alpha < +25°;\ \delta = 0;\ 0 < k' < +0.25$

$$\left|\begin{array}{l} \Delta C_Y = -0.060\ \alpha\ -0.7\ \alpha \left(\dfrac{k'}{2}\right)^2 \\ \Delta C_N = \ \ 0.053\ \alpha\ +0.8\ \alpha \left(\dfrac{k'}{2}\right)^2 \end{array}\right.$$

In the formulas, α and δ are expressed in radians.

The influence of k' on ΔC_Y and ΔC_N is small; the influence of δ, on the contrary, is relatively important. The effective force is always applied near the rudder; the distance from the center of gravity to the point of application is equal to about $0.90\ \frac{L}{2}$ while the distance from the center of gravity to the rudder stock is equal to $0.88\ \frac{L}{2}$.

6. FREE TURNING

The tests were made at 15-knot speed, on large models fitted with a propeller; a correction for friction was applied on the centerline of the model. Figure 3 gives the results (average between right and left turning).

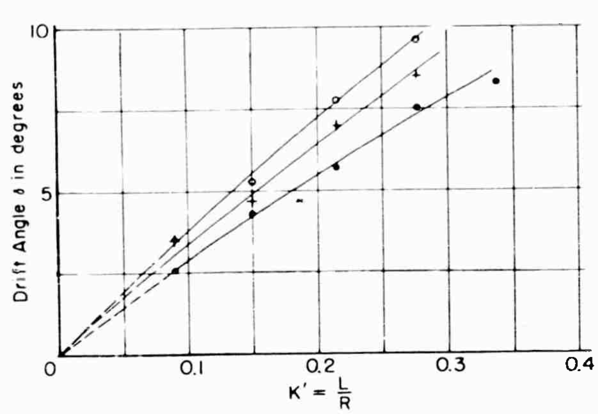


Figure 3a

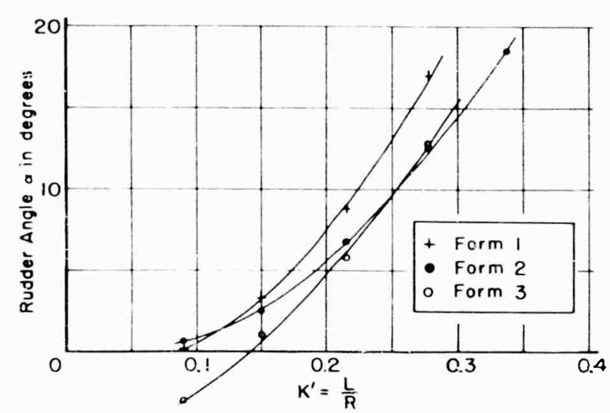


Figure 3b

Figure 3

For $\alpha = 0$, there exists a regime of free turning $k' = \frac{L}{R} \neq 0$. This result confirms that the straight-course regime is unstable.

Form 1 turns better than Form 2 (deeper) but with a larger drift angle.

III. CHARACTERISTIC COEFFICIENTS FOR A DESTROYER

1. FORM AND PROPORTIONS

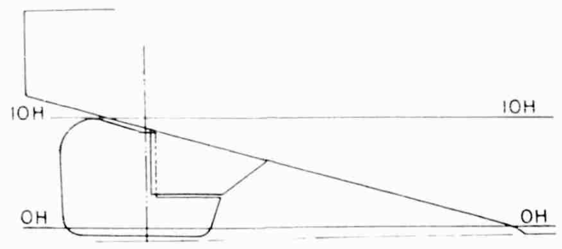
The characteristics of the full-scale ship are indicated in Table 4.

TABLE 4

Length (W.L.)	L = 123.50 m	
Breadth	B = 12.70 m	$\frac{B}{L} = 0.103$
Draft (midlength)	T = 3.90 m	$\frac{T}{L} = 0.0316$
Displacement	W = 3.200 m^3	$\frac{W}{LBT} = 0.523$
Maximum section area	$\otimes$ = 39.70 m^2	$\frac{\otimes}{BT} = 0.802$
Waterline area	Σ = 1225.40 m^2	$\frac{\Sigma}{LB} = 0.781$
Longitudinal area without appendages	S_0 = 445.40 m^2	$\frac{S_0}{LT} = 0.925$
Projected area of the rudder	S_1 = 16.40 m^2	$\frac{S_1}{S_0} = 0.0368$
Projected area of the horn	S_2 = 5.64 m^2	$\frac{S_2}{S_0} = 0.0127$
Distance from G to AP (W.L.)	ξ = 60.23 m	$\frac{\xi}{\frac{L}{2}} = 0.976$

The arrangement of the rudder and the horn is indicated in Figure 4.

Figure 4



2. TEST PROGRAM

Three forms have been tested:

Form 1 is geometrically similar to the full-scale ship;

Form 2 is obtained by expansion* of Form 1 draftwise: $\frac{T_2}{T_1} = 1.20$; and

Form 3 is obtained by expansion* of Form 1 beamwise: $\frac{B_3}{B_1} = 1.20$.

The models had a length of 2.50 m; they were fitted with turbulence pins.

The following tests have been done:

measurements of forces for $-10° \leq \delta \leq +10°$; $-0.3 < k' < +0.3$ without and with rudder (amidship) and horn; and

investigation of free turning.

All the tests have been carried out without propellers at a constant speed of 0.8 m/s.

3. CHARACTERISTIC TURNING COEFFICIENTS

Fairing has been done mechanically by the method of least squares. Measurements corresponding to too large angles of incidence have been discarded.**

The introduction of coefficients c_3 and c_3' have been judged necessary. On the contrary, the coefficients d_3 and d_3' have been taken, before fairing, equal to zero; the adjustment obtained with this assumption has been judged sufficient.†

The value of the ten characteristic coefficients are indicated in Table 5.

The presence of the rudder and of its horn has, for the three forms, a similar effect on each of the coefficients, except on a_1. The results obtained are not in contradiction with those of the cargo vessel; the variation of b_1 is, in particular, confirmed.

The change from Form 1 to Form 2 without or with rudder has a similar effect on each of the coefficients, except on c_3 and c_3'. The results obtained are not in contradiction with those of the cargo ship; variations of b_1 and b_3 are, in particular, confirmed.

The change from Form 1 to Form 3 without or with rudder has a similar effect on each of the coefficients, except on c_3, a_1' and c_3'. The increase of $\frac{B}{L}$ not having been studied for the cargo vessel, no comparison is possible between the two ships.

---

*Expansion of bare hull and rudder.

**Practically, for a left turn ($k'>0$), we have discarded the measurements made in the following conditions:

$$k' = 0.1,\ \delta \leq -10°;\ k' = 0.2,\ \delta \leq -8°;\ k' = 0.3,\ \delta \leq -6°.$$

†The quadratic mean of the errors $\frac{\sqrt{\Sigma e^2}}{n}$ was of the order of 15 grams for measurements amounting to about 500 grams.

TABLE 5

	Coefficients Relative to C_Y					Coefficients Relative to C_N				
	a_1	b_1*	a_3	b_3*	c_3	a'_1	b'_1	a'_3	b'_3	c'_3
Form 1 without rudder	0.85	−0.15 −1.32	21	−19 −18	18	0.85	−0.61	0.55	−20	2.6
with rudder and horn	0.82	0.21 −0.91	23	−14 −13	29	0.61	−0.80	−0.55	−23	0.1
Form 2 without rudder	0.61	0.00 −0.97	20	−14 −14	24	0.69	−0.49	1.50	−20	3.5
with rudder and horn	0.65	0.30 −0.63	20	− 9 − 9	28	0.55	−0.63	−0.20	−22	−3.4
Form 3 without rudder	0.61	−0.02 −1.43	15	−13 −12	20	0.71	−0.43	240	−18	2.7
with rudder and horn	0.76	0.16 −1.18	18	−12 −12	22	0.62	−0.68	0.55	−22	−1.3

* First Line: coefficient relative to C_Y (hydrodynamic).

Second Line: coefficient relative to C_Y total (hydrodynamic + centrifugal).

4. STABILITY OF STRAIGHT COURSE

Table 6 indicates the limiting positions T (towing with drift angle) and D (turning without drift angle).

TABLE 6

	$\frac{GT}{L/2}$	$\frac{GD}{L/2}$	$\frac{TD}{L/2}$
Form 1 without rudder	1.00	0.47	−0.53
with rudder and horn	0.74	0.88	+0.14
Form 2 without rudder	1.13	0.51	−0.62
with rudder and horn	0.85	1.00	+0.15
Form 3 without rudder	1.15	0.30	−0.85
with rudder and horn	0.82	0.58	−0.24

The influence of a skeg (rudder and horn) on stability on straight course is very clear; it draws back point T* and advances point D.** These two effects are favorable for straight-course stability.

Form 1 (normal) and Form 2 (deepened) are more stable than Form 3 (broadened). With rudder, Forms 1 and 2 are stable and Form 3 remains unstable.

5. FREE TURNING

Figure 5 gives the results of free turning tests made on the three forms.

Rudder at zero, there exists for Form 3 a turning regime possible corresponding to $k' = \frac{L}{R} = 1.4$; this result confirms that Form 3 is unstable.

For Forms 1 and 2, the curve $\left(\frac{L}{R}\ ;\ \alpha\right)$ is for $\frac{L}{R} < 0.1$ very near the abscissa axis: these two forms are then at the boundary between stability and instability; the tests of forced turning, which correspond to a number of important measurements show, however, that the stability is positive.

For rather large rudder angles, Form 1 turns poorer than Forms 2 and 3. Form 2 (deepened) turns as well as Form 3 (broadened) and with a smaller drift angle.

From the coefficients of Table 5 it is possible to calculate, for conditions k' and δ of the free turning, the forces which would act on the hull if the rudder were at zero; the normal force which results from deflecting the rudder is then deduced. The results are given, for $k' = 0.2$ and $k' = 0.3$ in Table 7.

TABLE 7

	k'	$\delta°$	$\alpha°$	C_Y	C_N	$\frac{C_N}{C_Y}$	$\frac{C_Y}{\alpha}$*
Form 1	0.2 0.3	6.5 9	7.4 20	0.052 0.139	−0.042 −0.113	0.81 0.82	0.40 0.40
Form 2	0.2 0.3	5.5 7.7	5.6 14.4	0.036 0.102	−0.034 −0.090	0.95 0.89	0.37 0.41
Form 3	0.2 0.3	7.3 10.1	4.7 15.3	0.025 0.089	−0.024 0.095	0.97 1.07	0.30 0.33

*α in radians

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* $\frac{GT \text{ with rudder}}{GT \text{ without rudder}} \sim 0.7$

** $\frac{GD \text{ with rudder}}{GD \text{ without rudder}} \sim 1.9$

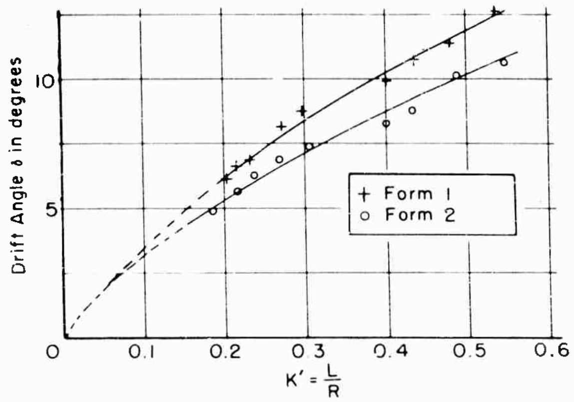

Figure 5a

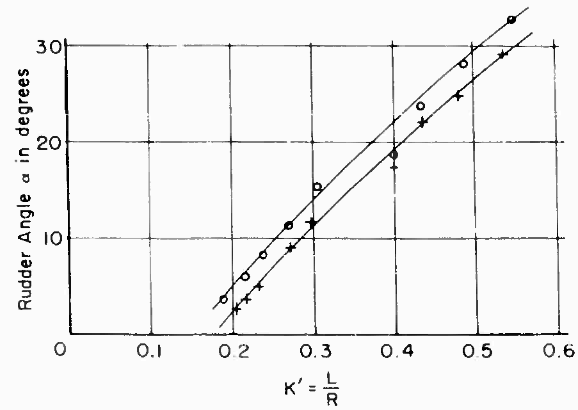

Figure 5b

Figure 5

From the standpoint of forces developed on the hull, the forms are classed in order 3, 2, 1, Form 3 (broad) being more favorable for turning. From the standpoint of the efficiency of the rudder, the forms are classed in the order 1 and 2, then 3.*

The combined action of the hull and of a clear rudder explains that the deep form, which is most satisfactory from the point of view of stability on straight course, gives results equivalent to those of the broad form for turning at rather large rudder angles.

IV. CONCLUSIONS

Regarding forms, proportions, and rudder and propeller arrangements, a cargo ship and a destroyer are the most different vessels we can imagine. It seems then very instructive to try to draw a parallel between the two ships. It will be the subject of our conclusions which recapitulate the principal results given in detail in Parts II and III.

1. CHARACTERISTIC COEFFICIENTS OF BARE HULLS

The bare hulls of the cargo ship and the destroyer have a similar behavior. The coefficients of the first order, which are important for motions in the neighborhood of straight-line regime, are of the same order of magnitude; the coefficients of the normal destroyer (Form 1) are, on the whole, nearer the coefficients of the cargo ship than those of the deepened and broadened destroyer.

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*If the rudder efficiency is taken as $\frac{C_{Y_1}}{\alpha}$, with $C_{Y_1} = \frac{Y}{\frac{1}{2}\rho S_1 V^2}$ the advantage of rudder 2 (largest aspect ratio) will be obvious since C_{Y_1} is proportional to $\frac{T}{L} C_Y$.

The difference between the two ships is chiefly appreciable on the coefficients of the third order which are important for turning with rather large rudder angles, but the combined influence of δ and k' gives for the two ships, forces of the same direction in the base of free turns.

Although both hulls are very different from a flat plate, it is possible to outline an approach to the theoretical method of Brard and Casal. In a straight course with a drift angle $\left(\delta \neq 0 \text{ but small; } k' = \frac{L}{R} = 0\right)$ the transverse force on the model has approximately the same position as that calculated $\left(GT = \frac{L}{2}\right)$; its value is smaller (20 to 40 percent than the calculated value ($C_Y = \delta$) for a flat plate. On the contrary, the hydrodynamic force due to turning ($\delta = 0$; $k' \neq 0$ but small) is much smaller than the calculated force $\left(C_Y = \frac{k'}{2}\right)$ for a flat plate, the measured moment being itself smaller (40 to 60 percent) than the calculated moment. These results are encouraging for comprehending phenomena by theoretical methods.

2. EFFECT OF A RUDDER (AMIDSHIP)

The presence of a stern skeg (rudder amidship) leads to similar effects for the cargo ship and the destroyer, but the orders of magnitude are not the same. One part of the difference comes from the fact that the skeg of the cargo ship is relatively less important than the skeg of the destroyer $\left(\frac{S_1}{S_0} = 0.0165 \text{ for the first; } \frac{S_1 + S_2}{S_0} = 0.0495 \text{ for the second}\right)$. The remainder comes from the nonidentity of the architectural arrangements. In the case of the cargo ship, the rudder is masked by the hull and its effect remains moderate when the propeller is not in place. On the contrary, in the case of the destroyer, the rudder and its horn, which are well clear from the hull, have an important effect: the limiting point T in straight course with a drift angle is brought back, whereas the limiting point D in turns without drift angle is advanced; finally, the stability of straight course is considerably improved by the presence of the rudder and its horn.

3. EFFECT OF CHANGE OF PROPORTIONS

For the cargo ship, only the ratio $\frac{T}{L}$ was changed, whereas for the destroyer $\frac{T}{L}$ and $\frac{B}{L}$ were successively changed. Changing $\frac{T}{L}$ has not led to contradicting results for the coefficients of the cargo ship and the destroyer. The deepening of the hull does not seem to have a very important effect on stability; on the contrary, the broadening of the hull has an unfavorable effect.

For the free turn, changing $\frac{T}{L}$ led to discrepant results for the cargo ship and the destroyer. In the case of the cargo ship the deepened model turns poorer than the normal model; it is the contrary in the case of the destroyer. In our opinion, the explanation of this diver-

gence is as follows: the increase of aspect ratio of an isolated rudder is favorable but this advantage remains behind the hull only if the rudder is well separated; it is exactly the case of the destroyer, whereas for the cargo ship the interactions between the hull and the rudder are large.

For the destroyer the deepened model turns as well as the broadened model whose stability on straight course is negative. It is confirmed here once more that stability on course and steady turning are different and not opposite qualities.

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DIRECTIONAL STABILITY OF AUTOMATICALLY STEERED SHIPS WITH PARTICULAR REFERENCE TO THEIR BAD PERFORMANCE IN ROUGH SEA

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INTRODUCTION

Use of an automatic steering device has been common practice for course-keeping at sea. The device, called an "autopilot," detects the deviation of a ship from a predetermined course and actuates a rudder so as to eliminate the deviation. The basic action of the autopilot is called "proportional control," which means to give a helm angle proportional to the amount of course deviation and sometimes also proportional to the time rate of the deviation. In this connection, a number of theoretical studies on automatic steering have been made since Minorsky first looked into this problem in 1922.[1,2,3] These studies also related largely to proportional control, with good success in linear treatments of the problem.

Proportional control must, however, be modified for practical application in a rough sea; otherwise, ceaseless alternating steering in response to each wave encountered would result, with no effect on course-keeping because of its rapid alternation. Such steering also might induce considerable wear of the steering gear and the autopilot itself. In these circumstances, "weather adjust" mechanisms have been used traditionally for avoiding the frequent steering. These mechanisms are composed of an amount of backlash (that is, idle clearance) inserted into linkages transmitting a course deviation signal, which decreases considerably the freqrent steering in a rough sea by neglecting those course deviations less than the amount of backlash. Unfortunately, however, the mechanisms, in transmitting the course deviation signal, also yield a phase lag which has so great an unstabilizing effect on ship motion as to induce sometimes a self-exciting, sustained yawing. According to a survey of the records of yawing of ships at sea, the well-known bad performances of autopilots in rough seas seem to result from this reason, and a loss of propulsive power caused by the yawing may exceed 10 percent.[4] This power loss is of course considerable; thus one of the most important subjects in steering problems of ships may be to analyze the self-exciting yawing resulting from conventional weather-adjust mechanisms, and to devise some new means of avoiding the frequent steering in rough seas without inducing such a harmful performance.

The present paper discusses these problems, utilizing a recent approach of analyzing nonlinear servomechanisms, and proposes a new principle of improving the autopiloting performance.

1. STABILITY CRITERION OF AUTOPILOTING BY KOCHENBURGER'S METHOD

A control loop describing autopiloting of a ship is illustrated in Figure 1 in the form of a block diagram. Since the stability criterion of Kochenburger[5] is based upon whether sinusoidal signals grow or decay in circulating through the loop, it is necessary first to obtain the response of each element composing the system to sinusoidal signals.

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[1]References are listed on page 356.

NOTE: The present paper is an English translation of the paper "Stability of Autopiloting," Journal, Society of Naval Architects of Japan, 1958, but with partial retouches.

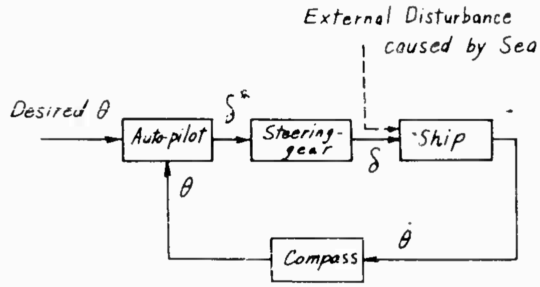

Figure 1 – Scheme of Automatic Steering of a Ship

The response of a ship to steering—or, in more traditional words, maneuverability of a ship—is usually described by a set of equations of motion which relates to a coupled motion of side-drifting and turning angular rotation. It is, however, more convenient to use a single equation of motion describing immediately a relation between turning angular motion and steering, because control signal in autopiloting relates to only turning angular motion. Such an equation of motion may be obtained by eliminating a drifting motion from the original simultaneous equations of motion, as follows:[6]

$$T_1 T_2 \frac{d^2\dot{\theta}}{dt^2} + (T_1 + T_2) \frac{d\dot{\theta}}{dt} + \dot{\theta} = K\delta + KT_3 \frac{d\delta}{dt} \qquad [1]$$

where $\dot{\theta}$ is the turning angular rate of a ship,

δ is the helm angle as a function of time, and

K, T_1, T_2 and T_3 are coefficients depending on hull forms, relative rudder size, and other factors of a ship.

The response of a ship to a sinusoidal signal (that is, in this case, to put a rudder sinusoidally to both sides in a certain frequency ω) may be determined through the equation: Namely,

$$\delta_{(t)} = \delta_0 \sin \omega t\,, \qquad [2]$$

$$\dot{\theta}_{(t)} = A_{(\omega)}\, \delta_0 \sin [\omega t + \phi_{(\omega)}],$$

where

$$A_{(\omega)} = K \sqrt{\frac{1 + \omega^2 T_3^2}{1 + \omega^2 (T_1^2 + T_2^2) + \omega^4 T_1^2 T_2^2}} = \left| \frac{K(1 + i\omega T_3)}{(1 + i\omega T_1)(1 + i\omega T_2)} \right|$$

$$\phi_{(\omega)} = -\tan^{-1} \frac{(T_1 + T_2 - T_3)\omega + T_1 T_2 T_3 \omega^3}{1 - (T_1 T_2 - T_2 T_3 - T_3 T_1)\omega^2} = \text{Arg}\, \frac{K(1 + i\omega T_3)}{(1 + i\omega T_1)(1 + i\omega T_2)}$$

and where δ_0 is the amplitude of sinusoidal steering. $A_{(\omega)}$ is called an amplitude ratio and $\phi_{(\omega)}$ a phase difference. Both of them are functions of frequency ω only, as a common feature of linear systems.

Next, the response of an electrohydraulic steering gear, which is widely used for most present ships, may be described by the following equation:

$$T_E \frac{d\delta}{dt} + \delta = \delta^*$$

where T_E is a time constant of a steering gear,

δ^* is a helm angle that is called for by the control, and

δ is a helm angle that is produced by a steering gear.

A synopsis of introducing the equation will be discussed in Section 3.2. Then we get a description of the response of a steering gear to a sinusoidal δ^* as follows:

when

$$\begin{aligned} \delta^* &= \delta_0^* \sin \omega t, \\ \delta &= A_{E(\omega)}\, \delta_0^* \sin [\omega t + \phi_{E(\omega)}] \end{aligned} \qquad [3]$$

where $A_{E(\omega)} = \left| \dfrac{1}{1 + i\omega T_E} \right|$ and $\phi_{E(\omega)} = \text{Arg}\ \dfrac{1}{1 + i\omega T_E}$.

Finally, we get easily the response character of a compass as follows, because it may be considered a simple integrating element transforming $\dot{\theta}$ into θ. Namely,

when

$$\begin{aligned} \dot{\theta} &= \dot{\theta}_0 \sin \omega t, \\ \theta &= A_{C(\omega)}\, \dot{\theta}_0 \sin [\omega t + \phi_{C(\omega)}] \end{aligned} \qquad [4]$$

where $A_{C(\omega)} = \left| \dfrac{1}{i\omega} \right| = \dfrac{1}{\omega}$ and $\phi_{C(\omega)} = \text{Arg}\ \dfrac{1}{i\omega} = -\dfrac{\pi}{2}$.

Then summing up the relations [2], [3], and [4], we know how a sinusoidal signal of δ^* is transformed in transmitting through a steering gear, a ship, and a compass successively. Namely, when

$$\begin{aligned} \delta^* &= \delta_0^* \sin \omega t, \\ \theta &= A_{(\omega)}\, A_{E(\omega)}\, A_{C(\omega)}\, \delta_0^* \sin [\omega t + \phi_{(\omega)} + \phi_{E(\omega)} + \phi_{C(\omega)}] \\ &\equiv A_{L(\omega)}\, \delta_0^* \sin [\omega t + \phi_{L(\omega)}] \end{aligned}$$

where

$$A_{L(\omega)} = A_{(\omega)}\, A_{E(\omega)}\, A_{C(\omega)} = \left| \frac{K\,(1 + i\omega T_3)}{i\omega\,(1 + i\omega T_1)(1 + i\omega T_2)\,(1 + i\omega T_E)} \right|$$

$$\phi_{L(\omega)} = \phi_{(\omega)} + \phi_{E(\omega)} + \phi_{C(\omega)} = \text{Arg} \frac{K(1+i\omega T_3)}{i\omega(1+i\omega T_1)(1+i\omega T_2)(1+i\omega T_E)}$$

It is convenient for the present treatment to rewrite the relation in a complex function expression as follows: when $\delta^* = \delta_0^* e^{i\omega t}$ (note that $e^{i\omega t} = \cos \omega t + i \sin \omega t$),

$$\theta = A_{L(\omega)} e^{i\phi_{L(\omega)}} \cdot \delta_0^* e^{i\omega t} \qquad [5]$$

Then a complex function $A_L e^{i\phi_L}$ represents an overall response character of all linear elements composed of a ship, steering gear, and compass.

Since the transmitting character through all linear elements has been thus obtained, if a similar character of the remaining element, viz, an autopilot, is defined, we can judge whether sinusoidal signals grow or decay in circulating through the control loop. It is impossible, however, to describe the response of an autopilot by any linear differential equation and then to obtain its response character in the foregoing manner, if considering such a discontinuous (then naturally nonlinear) element as a weather-adjust mechanism, which is one of the major objects of the present treatment.

Fortunately, however, the response itself of an autopilot with a weather-adjust mechanism to a sinusoidal input signal (that is, in this case a course deviation) may be obtained by an easy reasoning, as is indicated in Figure 2 and the formulas [7]. Then we expand the response into a Fourier series with a fundamental frequency that is the same with the frequency of the input signal, as follows:

$$\delta^* = C_1 \theta_0 (a_1 \sin \omega t + a_2 \cos \omega t + a_3 \sin 2\omega t + a_4 \cos 2\omega t + \ldots)$$

where C_1 is the proportionality constant connecting a course deviation to a helm angle to be called for,

θ_0 is the amplitude of course deviation, and

a_1, a_2, a_3, and a_4 are the coefficients of Fourier series which may be obtained through the usual procedure of Fourier expansion if the form of δ^* is given.

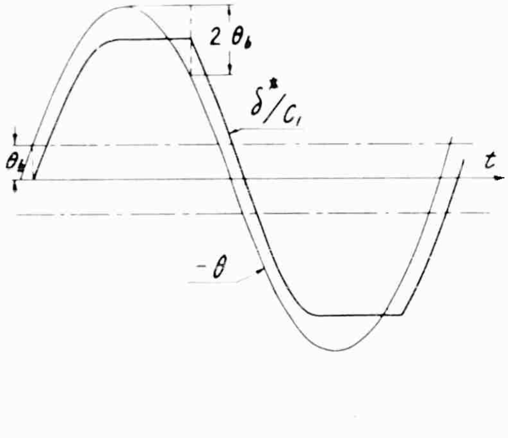

Figure 2 – Action of Backlash Weather-Adjust

Considering here that all linear elements (viz, a ship, steering gear, and compass) are much more insensitive to a signal with the higher frequency, we can neglect all the higher frequency terms. This is the approximation of Kochenburger, and its validity depends upon how much the higher frequency signal decays through the linear elements and also how much higher frequency components are included in the original δ^*; viz, how much the original δ^* resembles a pure sinusoidal form. In the present case, Kochenburger's approximation may be fairly valid because a ship is quite insensitive to a high-frequency steering because of her large inertia, and also the basic action of an autopilot, even with a weather-adjust, is proportional control that produces a sinusoidal output in response to a sinusoidal input. This is shown clearly in Figure 6, which illustrates a record of yawing of a full-loaded cargo boat in rough sea under autopiloting.[4] Although this autopilot has a weather-adjust mechanism and another discontinuous element called a telemotor-adjust (refer to Section 2.3), the resulting ship motion may be considered a sinusoidal motion, on the whole, except for small ripples caused by each wave encountered.

Thus, neglecting higher frequency terms in the output of an autopilot, we obtain a description of a response character of an autopilot even with a nonlinear element in a similar form as for a linear element, as follows:

when $\theta = \theta_0 \sin \omega t,$

$$\delta^* = C_1 A_N \theta_0 \sin (\omega t + \phi_N)$$

where $A_N = \sqrt{a_1^2 + a_2^2}$ and $\phi_N = \tan^{-1} \dfrac{a_2}{a_1}$.

It should be carefully noted that the amplitude ratio A_N and phase difference ϕ_N are functions not of a frequency ω as for linear elements but of an input amplitude θ_0 in this case, because Fourier coefficients a_1 and a_2 vary naturally with forms of δ^* and then with amounts of θ_0. This is just one of the remarkable features of nonlinear elements.

Rewriting the relation in the complex form, we get

$$\delta^* = A_N e^{i\phi_N} \theta_0 e^{i\omega t} \quad \text{when} \quad \theta = \theta_0 e^{i\omega t} \qquad [6]$$

The complex function $A_N e^{i\phi_N}$ is called a describing function of the nonlinear element.

Then considering together the relations [5] and [6], we can determine how the amplitude and phase of a sinusoidal signal vary in circulating through the control loop; the amplitude is multiplied by $A_L A_N$ and the phase shifts by $(\phi_L + \phi_N)$. The overall amplitude ratio $A_L A_N$ and phase shift $(\phi_L + \phi_N)$ are functions of the frequency and initial amplitude of a circulating signal.

Now assuming an amount of initial amplitude θ_0, the overall phase shift may equal 2π for a certain ω. If the overall amplitude ratio $A_L A_N > 1$ for these assumed θ_0 and ω, a signal with θ_0 and ω may grow up in circulating through the loop because a phase shift of 2π is

identical with no phase shift. Then in this case the control system is unstable for the initial amplitude θ_0. On the contrary, if $A_L A_N < 1$ for these θ_0 and ω, the signal may decay in circulating through and then the system is stable for the initial amplitude θ_0. This is the principle of Kochenburger's stability criterion. This criterion is usually carried out in a graphic procedure of the following manner (referring to Figure 3):

1. Plot $A_N e^{i\phi_N}$ and $1/A_L e^{i\phi_L}$ on a complex variable plane, taking a parameter as θ_0 for the former and as ω for the latter;

2. If the curve representing $1/A_L e^{i\phi_L}$ encircles completely the other curve representing $A_N e^{i\phi_N}$, the control system is stable for all conditions. This is the case of the chain lines in Figure 3.

3. If the two curves intersect, the system is stable for those amplitudes for which the $A_N e^{i\phi_N}$ curve is inside the $1/A_L e^{i\phi_L}$ curve, and unstable for those amplitudes for which the $A_N e^{i\phi_N}$ curve is outside the $1/A_L e^{i\phi_L}$ curve.

Then there may be two cases: one where the system is unstable in small amplitudes and stable in large ones, and the other where the system is stable in small amplitudes and unstable in large ones. In the former case, a self-exciting oscillation occurs because the system is unstable at rest, and the oscillation grows up to an amplitude represented by the intersection between the two curves because the system becomes stable for the larger amplitudes. Thus finally appears a sustained oscillation with an amplitude and frequency represented by the intersection. This is so-called "soft self-excitation," and a sustained

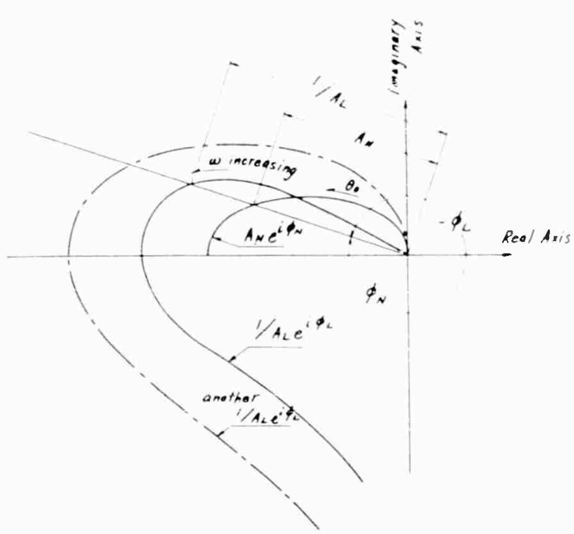

Figure 3 – Illustration of Kochenburger's Stability Criterion

yawing of a ship under autopiloting may be accounted as this kind of oscillation, as discussed in Section 2.2. The latter case corresponds to so-called "hard self-excitation," but it has no relation to the present analysis.

This graphic procedure is based upon the foregoing principle of the Kochenburger criterion. It may be easily understood by noting that a set of cross points where a radial line from the origin intersects the two curves represents those combinations of θ_0 and ω for which the overall phase shift $(\phi_L + \phi_N)$ equals 2π, and also that the $1/A_L e^{i\phi_N}$ curve outside the $A_N e^{i\phi_N}$ curve means that $1/A_L > A_N$ and then $A_L A_N < 1$.

2. SUSTAINED YAWING CAUSED BY WEATHER-ADJUST ACTION, AND A SURVEY OF SEVERAL MEANS FOR AVOIDING SUSTAINED YAWING

2.1 FUNCTION OF A WEATHER-ADJUST MECHANISM

A weather-adjust action has been provided traditionally by inserting a backlash (that is, an idle clearance) into linkages transmitting a course-deviation signal. Now denoting the amount of the clearance converted into the scale of course deviation by $2\theta_b$, a response of an autopilot with a weather-adjust to a sinusoidal input signal $\theta = \theta_0 \sin \omega t$ may be obtained as follows by reasoning of the action of the autopilot referring to Figure 2.

$$
\begin{aligned}
&\text{for} \quad -\frac{\pi}{2} + \cos^{-1}\left(1 - \frac{2\theta_b}{\theta_0}\right) \leqq \omega t \leqq \frac{\pi}{2} & \frac{\delta^*}{C_1} &= -\theta_0 \sin \omega t - \theta_b \\
&\frac{\pi}{2} \leqq \omega t \leqq \cos^{-1}\left(1 - \frac{2\theta_b}{\theta_0}\right) + \frac{\pi}{2} & : &= -\theta_0 - \theta_b \\
&\cos^{-1}\left(1 - \frac{2\theta_b}{\theta_0}\right) + \frac{\pi}{2} \leqq \omega t \leqq \frac{3}{2}\pi & : &= -\theta_0 \sin \omega t + \theta_b \\
&\frac{3}{2}\pi \leqq \omega t \leqq \frac{3}{2}\pi + \cos^{-1}\left(1 - \frac{2\theta_b}{\theta_0}\right) & : &= \theta_0 + \theta_b
\end{aligned}
\qquad [7]
$$

where C_1 is a constant of proportional control connecting δ^* to course deviation θ.

Although recently some new kinds of autopilots transmitting control signals not by mechanical linkages but by electric circuits are used more widely, the action described by the formulas [7] has been followed in these new devices.

Then following Kochenburger, we expand this δ^* in a Fourier series and take the fundamental frequency components only.

$$\delta^* = C_1 \theta_0 (a_1 \sin \omega t + a_2 \cos \omega t)$$

$$a_1 = -\left\{1 - \frac{\cos(1-2b)}{\pi}\right\} - \frac{2}{\pi}(1-2b)\sqrt{b(1-b)}$$

$$a_2 = \frac{4b}{\pi}(1-b)$$

where
$$b = \frac{\theta_b}{\theta_0} .$$

Plotting of $A_N e^{i\phi_N}$ is made by putting a_1 along the real axis and a_2 as the imaginary axis. Then we get a nearly elliptic curve, as shown in Figures 4 and 5. The constant C_1 may be included in linear elements by multiplying A_L by C_1.

2.2 SELF-EXCITING YAWING CAUSED BY WEATHER-ADJUST

Taking three representative ships as indicated in Figures 4 and 5, we examine directional stability under autopiloting. These ships may be considered to represent well the merchant ships in use at present so far as maneuverability is concerned. Steering gears in the treatment are usual electrohydraulic ones with a steering rate of hard-over to hard-over per 30 sec. The constant of proportional control C_1 is varied as 1, 2, and 4.

The results indicate that the two ships in full-load condition get into self-exciting, sustained yawing by weather-adjust, except for one case of the cargo boat (Ship B) with the smallest C_1 because the $1/A_L e^{i\phi_L}$ curve is inside the $A_N e^{i\phi_N}$ curve for small amplitude (large b) and the former outside the latter for large amplitude (small b). The amplitude and frequency of the yawing depend on the dynamic character of a ship represented by the indices K, T_1, T_2, and T_3 and also the constant of proportional control C_1. The amplitude is also

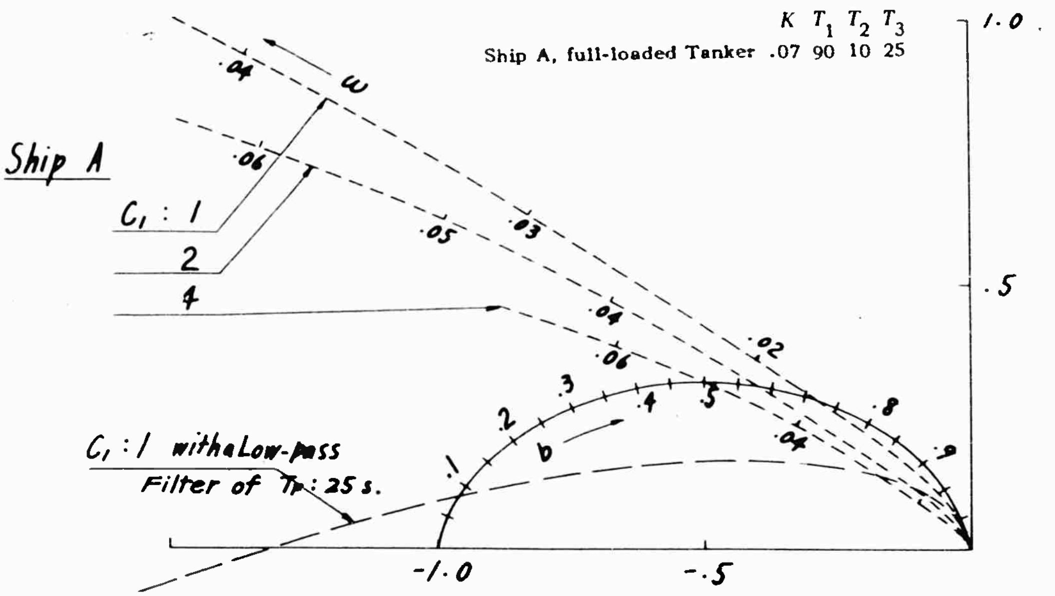

Figure 4 – Stability Criterion for Autopiloting of a Full-Loaded Tanker, Showing Self-Exciting Yawing

	K	T_1	T_2	T_3
Ship B, full-loaded Cargo-boat	.08	45	6.0	10
Ship C, ballasted Cargo-boat	.06	12	2.0	5.0

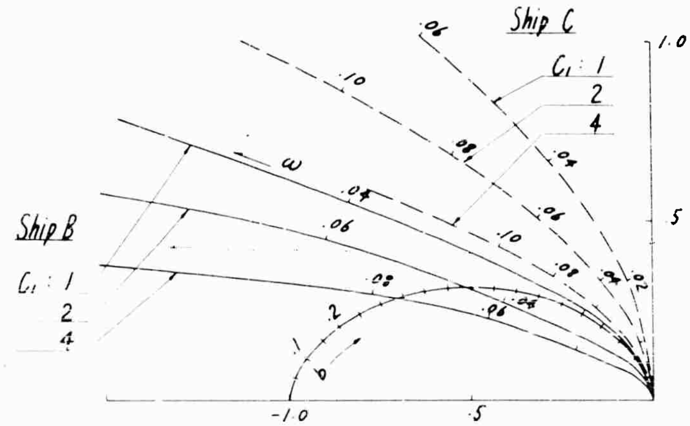

Figure 5 – Stability Criterion for Autopiloting of a Full-Loaded Cargo Boat and a Ballasted Cargo Boat

proportional to the amount of weather-adjust θ_b, and it rises in this case to 1.5~2.5 θ_b. The period distributes from several scores of seconds to 200~300 sec at the largest. This yawing is naturally accompanied with a self-exciting alternating movement of a rudder, and its amplitude is C_1 times as large as the amplitude of yawing.

Yawing records of a full-loaded cargo boat under autopiloting with weather-adjust obtained by Motora,[4] a sample of which is shown in Figure 6, indicate such an order of

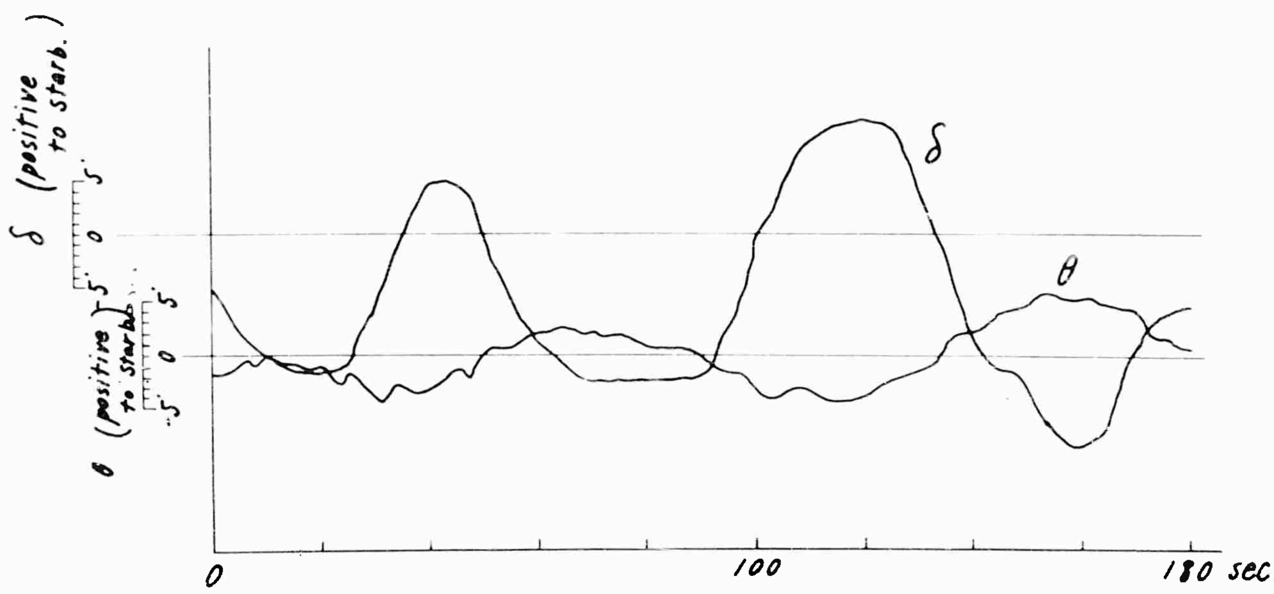

Figure 6 – Yawing Record of a Full-Loaded Cargo Boat under Autopiloting in Rough Sea

sustained yawing, and it may result from this reason. According to Motora, a loss of propulsive power caused by the yawing may rise to 10 ~ 20 percent of a normal one. This is of course quite a considerable power loss.

A ballasted cargo boat (Figure 5) does not get into the self-exciting yawing even for the largest proportional control constant; viz, $C_1 = 4$. This indicates that those ships with excellent inherent stability on course are free from the sustained yawing caused by weather-adjust.

2.3 MEANS OF AVOIDING SUSTAINED YAWING CAUSED BY WEATHER-ADJUST

Considering the foregoing severe power loss caused by sustained yawing and also other possible disadvantages of the accompanying wild steering, it is quite important to devise some effective means of avoiding the wild performance.

1. Weather-Adjust Using a Dead-Band Action

To use a "dead band" action in place of a backlash for weather-adjust may be one of the most preferable means for the present purpose. This action is indicated in Figure 7.

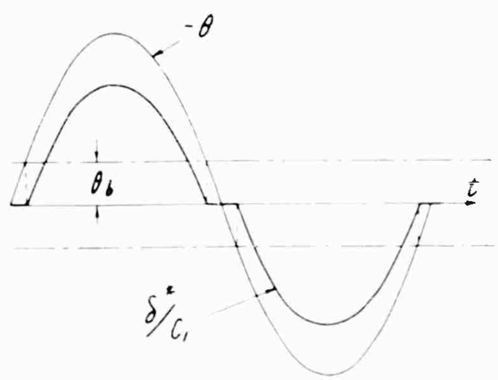

Figure 7 – Action of Dead-Band Weather-Adjust

Namely, the dead-band mechanism shows no response to an input signal smaller than a certain amount, for a signal larger than this amount the dead-band mechanism produces an output signal, the magnitude of which is always smaller than that of the input by the same amount, whether the input is increasing or decreasing. Then dead-band action induces no phase shift, and this is the important difference between dead-band and backlash.

Since the dead-band induces no phase shift for all amounts of input amplitudes, the $A_N e^{i\phi_N}$ curve for it is naturally a straight line along the real axis connecting the point (−1,0) to the origin. Then all possible $1/A_L e^{i\phi_L}$ curves are always outside the $A_N e^{i\phi_N}$ curve and therefore a ship under autopiloting with a dead-band mechanism in place of the usual weather-adjust backlash never gets into any sustained yawing.

Concerning the avoidance of frequent steering in response to each wave encountered, dead-band and backlash have quite the same performance. Thus to use dead-band as a new

weather-adjust mechanism may be one of the most preferable means for avoiding the frequent steering in a rough sea without inducing the sustained yawing.

There may be several practical devices of realizing the dead-band action by being inserted into the autopilot circuit; for instance, to short-circuit a certain width of the winding of a potentiometer generating a course deviation signal, or to use auxiliary relays for the purpose. The details of the device, however, remain with future development in autopilot design.

2. Weather-Adjust Using a Low-Pass Filter

The use of a low-pass filter for avoiding frequent steering in rough sea has been proposed for the first time by Motora,[4] and discussed in detail by Rydill.[7] The idea is based upon the fact that the frequency of each wave encountered is fairly high and therefore frequent steering may be avoided by inserting a circuit insensitive to high-frequency signal into a control loop of an autopilot. There is, however, a difficulty that such a low-pass filter induces generally a certain amount of phase shift in transmitting control signals and then spoils more or less the stability of the control system, also as is pointed out by Rydill. Thus the major subject concerning the use of a low-pass filter for the purpose is how to select parameters of a filter in order to obtain the most effective avoidance of frequent steering with the least loss of the stability of the whole system. This is a typical compromising problem and, employing a modern approach of synthesizing servomechanisms, Rydill has treated this problem.[7]

A long dotted curve in Figure 4 represents the $1/A_L e^{i\phi_L}$ curve for Ship A with a simple first-order low-pass filter with a time constant of 25 sec. Since in this case the weather-adjust backlash is replaced by the filter, the whole system stability may be judged from that distance at which the $1/A_L e^{i\phi_L}$ curve passes near the point (–1, 0) according to the Nyquist stability criterion that is also quoted in References 2 and 7. In this case, the phase margin (that is, an index of system stability in the Nyquist criterion) is very small (about 5 deg) and accordingly the stability is very low. On the other hand, the time constant of 25 sec is not sufficient for avoiding a steering in response to each wave, as a period of encounter may fairly exceed 10 sec in a following sea. Then in this case the use of the low-pass filter seems not so promising.

Although the second-order low-pass filter treated by Rydill indicates a more excellent performance for the purpose, there may remain a question of whether avoidance of the frequent steering in rough sea and the system stability may both be sufficient, even with the well-synthesized filter, particularly for those ships with not so good inherent stability on course, as are the cases for full-loaded usual merchant ships.

3. Use of Rate Control or Angular Velocity Control

A rate control means to actuate a rudder not only in proportion to course deviation but also to the time rate of the deviation. Namely, the control may be represented by the formula as follows:

$$\delta^* = -C_1\theta - C_2\dot{\theta}$$

This control is well known as the most effective means of improving the stability under autopiloting. Now let us consider the contribution of the rate control to eliminate the self-exciting yawing.

We take, as a standard extent of the rate control, so-called "optimum damping." Too much rate control results in a slow motion in course-changing, whereas too little rate control cannot show well enough the advantage of the control. An optimum damping is provided by adding just a sufficient and necessary extent of rate-control, and the extent may be easily estimated using the Nyquist criterion; viz, select the extent of rate-control so that a phase margin for that extent equals 35 deg ~ 40 deg. For instance, the optimum damping is provided for Ship A, a full-loaded tanker, by $C_2 = 3$ sec; and for Ship B, a full-loaded cargo boat, by $C_2 = 5 \sim 7$ sec, where C_2 represents a constant of rate control, indicated above.

Now carrying out Kochenburger's criterion for Ship A under autopiloting with the usual weather-adjust action and the optimum rate control, we know the ship still gets into a sustained yawing but with a smaller amplitude than for no rate control. The amplitude cannot be considered sufficiently small as the reduction is about 30 percent. Referring to the result, much more extent of rate control might be required in order to eliminate the yawing sufficiently by only this means. It seems that this is not a reasonable means of avoiding the sustained yawing.

4. Damping Using a "Negative" Backlash – A Telemotor-Adjust Mechanism

This mechanism has been used for a long time, together with the usual weather-adjust mechanism, and in some cases even new mechanisms employing an auxiliary relay to simulate this action are in use. In this mechanism an amount of backlash is inserted into a mechanical feedback linkage, which transmits an output signal δ^* back to an electric contact actuating a driving motor. The action of the mechanism is to compensate a phase lag in transmitting a control signal by "rewinding" that occurs when the input signal changes a direction of motion, as is shown in Figure 8. Although the original utility of the mechanism is to compensate a possible idle movement in a telemotor link, a stabilizing effect by the phase compensation is more important, and usually this backlash is adjusted so as to overcompensate an idle movement in a telemotor link to yield a "phase lead." Such an adjust is called "over-telemotor adjust." In other words, this mechanism is another kind of damping means to raise stability in autopiloting.

The stabilizing effect of the mechanism may be also analyzed by Kochenburger's criterion. Considering the mechanism together with a usual weather-adjust mechanism, as is shown in Figure 8, we obtain Kochenburger's description on a response of an autopilot with these mechanisms as follows:

when

$$\theta = \theta_0 \sin \omega t,$$

$$\frac{\delta^*}{C_1 \theta_0} = a_1 \sin \omega t + a_2 \cos \omega t$$

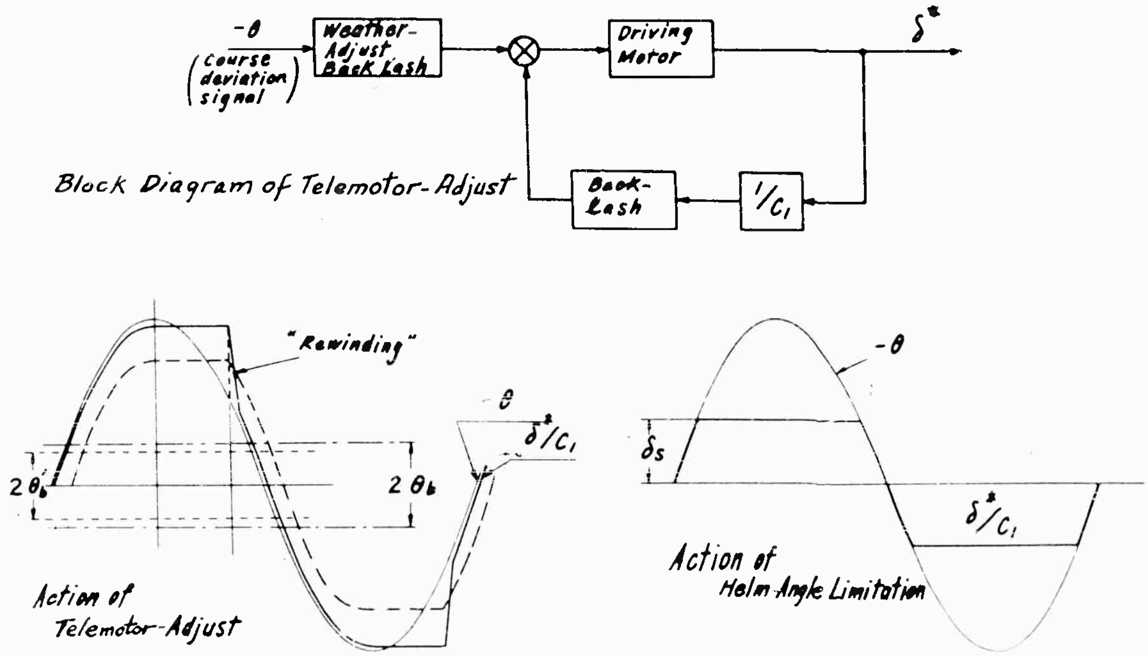

Figure 8 – Telemotor-Adjust and Helm-Angle Limitation

$$a_1 = -\left(1 - \frac{\cos^{-1}(1-2b)}{\pi}\right) - \frac{2}{\pi}\sqrt{b(1-b)} \cdot \{1 - 2b\,(1-2\beta)\}$$

$$a_2 = \frac{4}{\pi}\,\{b\,(1-b) - \beta b\,(1-2b)\}$$

where $b = \theta_b/\theta_0$ and $\beta = \theta_b'/\theta_b$ and where $2\theta_b'$ is the amount of "negative" backlash converted into the scale of course deviation. Kochenburger's plotting in this case is shown in Figure 9. It indicates that the amplitude of the sustained yawing decreases by about 20 percent through using this damping means with $\beta = 0.6$.

Since this amplitude is not sufficiently small and any more improvement cannot be expected by the mechanism referring to the result, it may be concluded that the damping by over-telemotor-adjust is not satisfactory to avoid the self-exciting yawing.

5. Helm-Angle Limitation

Most autopilots have helm-angle limitation mechanisms, the aim of which may be to avoid wild steering of a large amplitude experienced in rough sea. Denoting a limit angle of helm by δ_s, we obtain Kochenburger's description of the response of an autopilot with a helm-angle limitation, as well as a normal weather-adjust, as follows:

when $$\theta = \theta_0 \sin \omega t$$

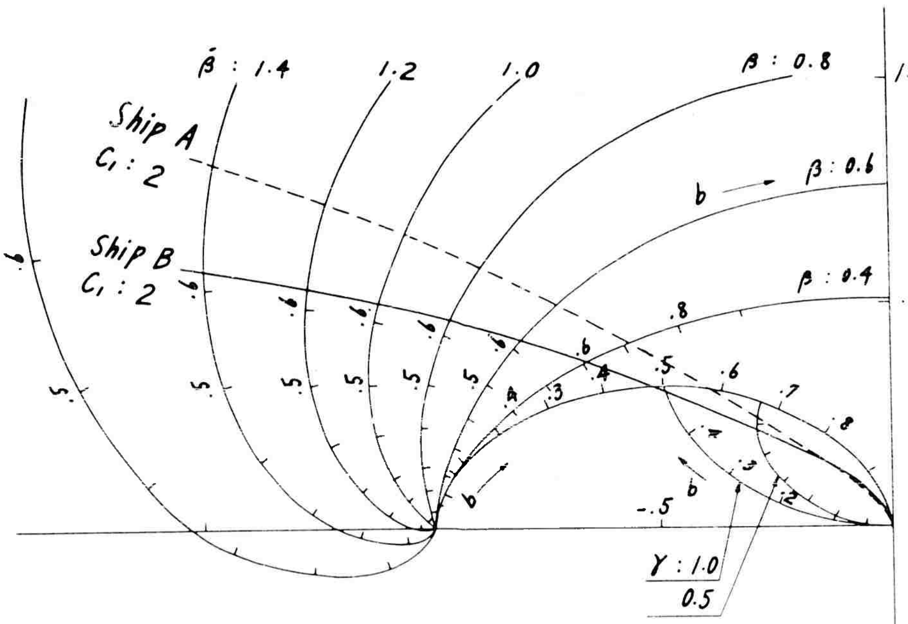

Figure 9 – Effects of "Over-Telemotor Adjust" Damping and Helm-Angle Limitation

$$\frac{\delta^*}{C_1\theta_0} = a_1 \sin \omega t + a_2 \cos \omega t$$

$$a_1 = -\frac{1}{\pi}\left[\cos^{-1}\{-(\gamma+1)b\} - \cos^{-1}(\gamma-1)b + b(\gamma+1)\sqrt{1-b^2(\gamma+1)^2} + b(\gamma-1)\sqrt{1-b^2(\gamma-1)^2}\right]$$

$$a_2 = \frac{4}{\pi}\gamma b^2$$

where $b = \theta_b/\theta_0$ and $\gamma = \dfrac{\delta_s}{C_1\theta_b}$. These formulas are valid for $b(\gamma + 1) \leqq 1$ and for $b(\gamma + 1) \geqq 1$, the helm-angle limitation has no effect, because it means that an amplitude of δ^* does not reach the limiting angle of helm. Koehenburger's plotting for this case is also shown in Figure 9 by long dotted lines; the helm limitation shows no effect for amplitudes smaller than $b(\gamma + 1) > 1$, and for amplitudes larger than $b(\gamma + 1) \leqq 1$ the $A_N e^{i\phi_N}$ curve returns to the origin. The result shows that the helm limitation of 10 deg somewhat reduces the frequency of the sustained yawing, while it does not affect the amplitude. If the limiting angle δ_s is reduced more, the slower is the sustained yawing (the smaller ω) and also the amplitude may decrease. It is difficult, however, to take a too small δ_s, because at least a certain amount of helm angle is required to keep a course against a sustained disturbance; for instance, a constant wind blowing abeam. Considering that the least necessary helm angle may rise fairly to 10 deg, it seems that the helm limitation is not so promising to ease the self-exciting yawing.

3. SOME NOTES ON EFFECTS OF INHERENT CHARACTERS OF A SHIP AND A STEERING GEAR UPON THE WHOLE SYSTEM PERFORMANCE

3.1 INHERENT MANEUVERABILITY OF A SHIP AND THE STABILITY OF THE WHOLE SYSTEM

An inherent character of a ship in steering, usually called steering quality or maneuverability, may be described by the equation of motion [1] or the transfer function in steering as follows:[6]

$$Y_{s(p)} = \frac{K\,(1+T_3p)}{(1+T_1p)\,(1+T_2p)}$$

The results given in Section 2.3 indicate that those ships with low inherent stability, that means large value of T_1, yield a low stability under autopiloting and then are easy to get into a self-exciting yawing. This is a reasonable conclusion.

Next the contribution of the index T_3 is worthy of notice. T_3 represents originally a contribution of steering rate in initiating turning motion and accordingly a large T_3 contributes largely to an inherent maneuverability. The effect of T_3 is, however, not so dominant that it is a fairly sufficient treatment to consider T_3, together with T_2, in a form of correction to T_1; viz, to use a single time constant $T = T_1 + T_2 - T_3$ in place of T_1 and to neglect T_2 and T_3. This is the first-order simulation which indicates wide utilities in a brief and plain description of steering motion of a ship.[6]

In the case of autopiloting, however, the effect of T_3 is quite dominant. Consequently the first-order simulation loses its validity in this case, as is shown in Figure 10. This depends on the fact that the frequency which is the most significant for the stability under autopiloting is relatively high, while the first-order simulation has originally its main utility in a low-frequency phenomenon.

3.2 RESPONSE CHARACTER OF ELECTROHYDRAULIC STEERING GEARS

A variable-displacement pump used for an electrohydraulic steering gear drives a ram, which is connected to a tiller by a speed that is proportional to $(\delta^* - \delta)$, where δ is the actual helm angle and δ^* is the helm angle called for. Then we obtain an equation of motion of a steering gear as follows:

$$T_E\,\frac{d\delta}{dt} + \delta = \delta^*$$

where T_E is the time constant of a steering gear that is related immediately to the foregoing proportionality constant connecting $(\delta^* - \delta)$ to the ram speed.

The time constant T_E has a value of about 1.6 ~ 1.8 sec for the usual steering gear with a steering rate of hard-over to hard-over per 30 sec. The inertia of all moving parts may be neglected because of the relatively slow movement. In addition, the above equation is

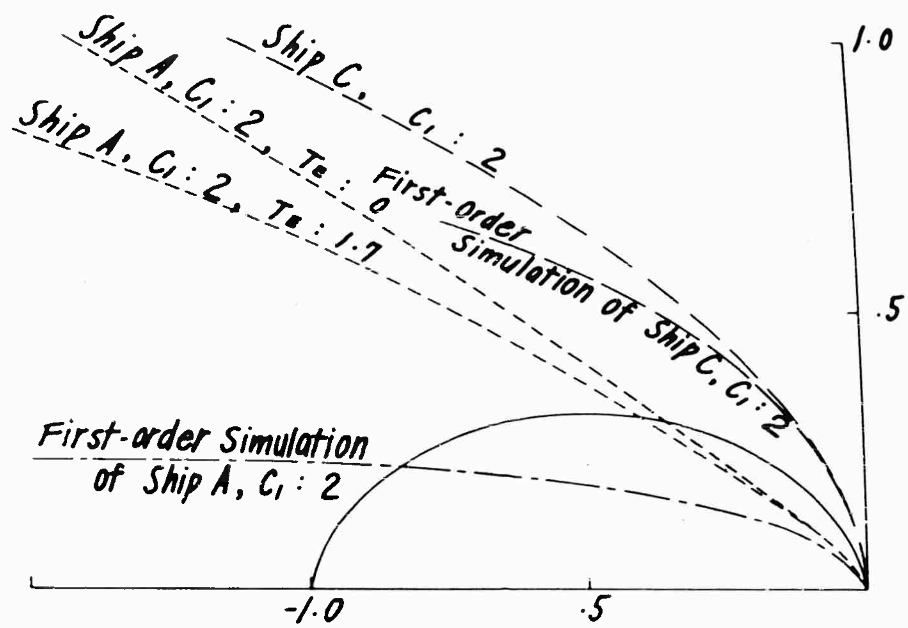

Figure 10 – Error of First-Order Simulation in Autopiloting Analysis and Effect of Steering Gear's Response Character

disturbed when $(\delta^* - \delta)$ fairly exceeds several degrees as the case of hard-over steering, but such a case does not appear usually under autopiloting.

To examine the effect of a response character of a steering gear, the $1/A_L \epsilon^{i\phi_L}$ curves with $T_E = 0$ and $T_E = 1.7$, that is a usual value, are compared in Figure 10. Referring to the result, a response character of a steering gear has slight effect upon the whole system performance, particularly for usual merchant ships, by reason of the relatively small value of its time constant.

CONCLUSIONS

1. The weather-adjust mechanism now in use, which is for avoiding a ceaseless steering in response to each wave encountered in rough sea, often induces a self-exciting, sustained yawing. A propulsive power loss caused by the yawing may sometimes exceed 10 percent of normal power.

2. Use of a dead-band mechanism in place of a backlash, the one usually used, may be the most preferable means of avoiding the ceaseless steering in rough sea without inducing any reduction of the stability of the whole system and then any bad performance as the self-exciting yawing.

3. An adequate amount of rate control is desirable in order to yield a sufficient stability under autopiloting. An optimum proportionality constant of rate control depends on the

dimensions, speed, relative rudder size, and load condition of a ship. It may be about 3 ~ 7 sec for full-loaded merchant ships.

4. Thus performances of present autopilots may be much improved by using a dead-band element for avoiding the ceaseless steering in rough sea, together with an adequate amount of rate control.

5. A bad inherent stability of a ship on course (a large value of the stability index T_1) injures also the stability under autopiloting. The time constant T_3, another index of the steering quality,[6] contributes so considerably to stability under autopiloting that the first-order simulation in steering, which has wide utilities on treatments on usual steering motion,[6] is completely invalid for the analysis of automatic steering.

6. The response character of a steering gear affects the performance of autopiloting relatively little, particularly in cases of full-loaded merchant ships which have fairly large time constants.

7. Stabilizing by means of so-called "over-telemotor-adjust" mechanisms is not very promising for autopiloting in rough sea.

8. Helm-angle limitation seems to be unnecessary so far as autopiloting performance is concerned.

ACKNOWLEDGMENT

Acknowledgment is made of the help by the staff of Tokyo Keiki Seizosho Company, Hokushin Denki Seisakusho Company, and Kawasaki Dockyard Company.

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Nr 45 GÖTEBORG 1960

A STUDY OF COURSE KEEPING AND MANOEUVRING PERFORMANCE

BY

NILS H. NORRBIN

GUMPERTS FÖRLAG
GÖTEBORG

1. Introduction

In the course of the last fifteen years, the interest in and knowledge of the many aspects of steering and control of ships have rapidly grown among naval architects. Today the subject is recognized as one of the utmost practical importance, and new facilities for the experimental verification of theories and for the guidance of future design are established at many model basins.

At the Swedish State Shipbuilding Experimental Tank (SSPA) there is as yet no manoeuvring basin, but a complete instrumentation is available for different types of tests with radio-controlled models in a small lake near Gothenburg. For use on the towing carriage in the 850 ft. model basin, the equipment includes one old, twin strut, all-mechanical, three-component balance (for submerged models only) and one single strut, resistance-wire strain gauge, six-component balance for internal mounting, both designed for the measurement of forces on captive models in stationary oblique towing. An exhaustive test program has been initiated for studies of a modern cargo liner form.

The reasons underlying the new inquiries are many. A constant stimulation is provided by the advancements in aerodynamics and control engineering and also by the new measuring techniques developed in all branches of ship model testing.

In this report some of the problems and progress are reviewed and an introduction is given to the modern treatment of the characteristics and stability of surface ship motion.

Section 3 contains a short attempt to summarize the achievements of the last eighty years, and the accompanying bibliography in Section 11 lists about ninety titles.

In Section 4 are formulated the general equations of motion for a body floating on the surface or submerged. Before proceeding to a further investigation of the stability and motion of a surface ship on a calm sea, two sections are devoted to a discussion of the stationary and non-stationary forces on hull and rudder, with illustrative examples.

The equations for the motion of a ship with a proportional rudder

control are simplified and normalized in Section 7. The next section leads to the algebraic criterion for inherent dynamic stability, deduced by inspection of the characteristic equation, whereas Section 9 makes use of elementary frequency response methods and the NYQUIST criterion for the directional stability of the steered ship.

The report ends with a few comments on manoeuvring performance related to course stability.

Gothenburg in September 1959.

2. Symbols and Units

When applicable, the symbols have been chosen in accordance with the nomenclature of the *Technical and Research Bulletin No. 1—5*, published by the Society of Naval Architects and Marine Engineers in 1950. In a few other cases, resort has been made to accepted practice in aerodynamics or control engineering.

In the discussion of component motions and of the hydrodynamic forces acting on the ship and its rudder, use is made of an orthogonal right-handed system of body axes, ($Gxyz$), with its origin in the centre of gravity and moving with the body. (Contrary to submarine standard code, a left-handed system of body axes is often used for surface ships, giving equal signs to the angle of drift and the corresponding drifting velocity.) The equations of motion are referred to these coordinates as well, the changes of linear and angular momentum of course being expressed with reference to a system of axes fixed in space. Ultimately the treatment is greatly simplified by considering a motion in the horizontal xy-plane only.

Two sketches for reference are presented in Figs. 1 and 2.

Symbol	Definition	Physical Dimension	Non-Dimensional Form as Used
A, B, C, D, E	Coefficients of the characteristic equation	—	—
B	Beam	L	B/L
$C(k_1)$	THEODORSEN's function (*cf.* p. 32)	—	—
$\vec{F}$	External force (vectorial)	MLT^{-2}	—
F_{nL}	FROUDE number	—	$U/\sqrt{gL}$
G	Centre of gravity	—	—
$G(j\omega')$	Total open loop transfer function (frequency-response form)	—	—
$G(s)$	Transfer function (operator form)	—	—
$\vec{H}$	Angular momentum (vectorial)	ML^2T^{-1}	—
H	Draught	L	H/B, H/L
I_{xx}, I_{yy}, I_{zz}	Mass moment of inertia about the axes x, y and z resp.	ML^2	$I'_{zz}=m'k_L^2$
K	Rolling moment about the x-axis	ML^2T^{-2}	—
K	Static loop gain, open loop	—	—
L	Length of ship (on WL)	L	—
L	Lift of wing	MLT^{-2}	$C_L=L/\frac{\varrho}{2}U^2 S$

Symbol	Definition	Physical Dimension	Non-Dimensional Form as Used
M	Magnification factor of closed loop	—	—
$\vec{M}$	External moment (vectorial)	ML^2T^{-2}	—
M	Pitching moment about the y-axis	ML^2T^{-2}	—
N	Yawing moment about the z-axis	ML^2T^{-2}	$N' = N/\frac{\varrho}{2}U^2 S_L L$
N_β	Typical stiffness derivative ($N_\beta = -UN_v$)	ML^2T^{-2}	$N'_\beta = N_\beta/\frac{\varrho}{2}U^2 S_L L$
N_r	Typical moment-angular velocity (rotary) derivative	ML^2T^{-1}	$N'_r = N_r/\frac{\varrho}{2}U S_L L^2$
$N_{\dot r}$	Typical moment-angular acceleration derivative	ML^2	$N'_{\dot r} = N_{\dot r}/\frac{\varrho}{2}S_L L^3$
N_v	Typical moment-velocity derivative	MLT^{-1}	$N'_v = N_v/\frac{\varrho}{2}U S_L L$
$N_{\dot v}$	Typical moment-acceleration derivative	ML	$N'_{\dot v} = N_{\dot v}/\frac{\varrho}{2}S_L L^2$
R	ROUTH's discriminant		—
R	Magnification factor of transfer function $\frac{\psi}{\Delta}(j\omega')$		
R_c	Radius of turning circle	L	R_c/L
S	Area in general	L^2	
S_L	Lateral area of reference (here equal to LH)	L^2	—
T	Period of time	T	$T' = TU/L$
U	Velocity of origin of body axes relative to the fluid; speed of ship	LT^{-1}	
U_1	Velocity of rudder relative to the fluid	LT^{-1}	U_1/U
X	Hydrodynamic force on body along the x-axis	MLT^{-2}	
Y	D:o along the y-axis	MLT^{-2}	$Y' = Y/\frac{\varrho}{2}U^2 S_L$
Y_β	Typical stiffness derivative ($Y_\beta = -UY_r$)	MLT^{-2}	$Y'_\beta = Y_\beta/\frac{\varrho}{2}U^2 S_L$
Y_r	Typical force-angular velocity (rotary) derivative	MLT^{-1}	$Y'_r = Y_r/\frac{\varrho}{2}U S_L L$
$Y_{\dot r}$	Typical force-angular acceleration derivative	ML	$Y'_{\dot r} = Y_{\dot r}/\frac{\varrho}{2}S_L L^2$
Y_v	Typical force-velocity derivative	MT^{-1}	$Y'_v = Y_v/\frac{\varrho}{2}U S_L$

Symbol	Definition	Physical Dimension	Non-Dimensional Form as Used
$Y_{\dot{v}}$	Typical force-acceleration derivative	M	$Y'_{\dot{v}}=Y_{\dot{v}}/\frac{\varrho}{2}\,S_L\,L$
$Y(\delta)$	Force due to rudder deflection, on rudder and hull	MLT^{-2}	$Y'(\delta)=Y(\delta)/\frac{\varrho}{2}\,U^2S_L$
Y_δ	Derivative of $Y(\delta)$	MLT^{-2}	$Y'_\delta=Y_\delta/\frac{\varrho}{2}\,U^2\,S_L$
Z	Hydrodynamic force on body along the z-axis	MLT^{-2}	—
a	Distance of rudder axis aft of leading edge	L	a/c
a	A parameter, defined by eq. (9.6)	—	—
c	Chord of wing or rudder	L	—
f	A parameter, defined by eq. (6.7)	—	—
g	Acceleration of gravity	LT^{-2}	—
h	A parameter, defined by eq. (9.8)	—	—
$\hat{i}, \hat{j}, \hat{k}$	Unit vectors along the axes of x, y and z resp.	—	—
k	Reduced frequency of ship yawing $\left(k=\pi\frac{L}{U}\ \nu=\frac{\omega'}{2}\right)$	—	—
k_1	D:o of oscillating rudder $\left(k_1=\frac{c\omega_1}{2U_1}\right)$		—
k_L	Longitudinal radius of gyration of ship mass in non-dimensional form	—	—
k_x, k_y	Coefficients of accession to inertia in translation along x- and y-axes	—	—
k_x^0, k_y^0	D:o, corrected for free surface neglecting gravity	—	—
k_{zz}	Coefficient of accession to moment of inertia in yawing about the z-axis	—	—
k_{zz}^0	D:o, corrected for free surface neglecting gravity	—	—
m	Mass of ship	M	$m'=m/\frac{\varrho}{2}\,S_L\,L$
m_x, m_y	Virtual mass of ship in translation along the x- and y-axes ($m_y=m\,(1+k_y)$ or $m_y=m\,(1+k_y^0)$)	M	$m'_x=m_x/\frac{\varrho}{2}\,S_L\,L$ $m'_y=m_y/\frac{\varrho}{2}\,S_L\,L$
p_1, p_2	Real roots of a quadratic characteristic equation with supercritical damping ($p_1 < p_2$)	—	—
q_i	Real parts of complex roots of a characteristic equation	—	—

Symbol	Definition	Physical Dimension	Non-Dimensional Form as Used
q	Stagnation pressure $\left(q=\frac{\varrho}{2}U^2\right)$	$ML^{-1}T^{-2}$	—
r	Angular velocity of yaw	T^{-1}	$r'=\frac{rL}{U}$
s	A derivation symbol: $s=\frac{d}{dt'}$. Also a root of the characteristic equation	—	—
t	Time	T	$t'=\frac{U}{L}t$
$\bar{t}$	Time lag due to finite rudder speed	T	$\bar{t}'=\frac{U}{L}\bar{t}$
u	Speed, or small change of speed of (centre of gravity of) ship along the x-axis	LT^{-1}	u/U
u_0	Constant speed of ship along the x-axis	LT^{-1}	u_0/U
v, w	Speed of ship along the axes y and z resp.	LT^{-1}	v/U, w/U
$\bar{w}$	Normal velocity of oscillating aerofoil	LT^{-1}	—
x, y, z	Orthogonal coordinates of a right-handed system of body axes, moving with the ship	L	—
x	Coordinate along chord of aerofoil	L	—
V	Volume displacement of ship	L^3	—
Λ	Aspect ratio ($\Lambda=\text{span}^2/\text{area}$)	—	—
Λ	Tuning factor ($\Lambda=\omega'/\omega_n'$)	—	—
$\Phi\,(k_1)$	Lift function of oscillating aerofoil	—	
$\frac{\Psi}{\Delta}\,(j\omega')$	Transfer function from rudder deflection to yaw	—	—
$\vec{\Omega}$	Angular velocity of ship (vectorial)	T^{-1}	—
Ω_1	Angular velocity of wing, pivoting about the mid-chord point	T^{-1}	—
α	Angle of attack (cf. Fig. 1)	—	—
β	Angle of drift or sideslip ($\beta=-v/U$)	—	—
γ	Coefficient of heading error term in proportional rudder control	—	$\gamma'=\gamma$
δ	Rudder angle (deflection)	—	
δ^*	Rudder angle ordered by automatic control	—	—
δ_B	Block coefficient ($\delta_B=V/LBH$)	—	—
η	Lateral area coefficient (true lateral area/product HL)	—	—
λ	Relative distance of rudder (rudder axis) aft of G of ship	—	—
μ	Relative density of ship mass	—	—
μ	Magnification factor for rate of yaw amplitude	—	—

Symbol	Definition	Physical Dimension	Non-Dimensional Form as Used
ν	Frequency of ship yawing oscillations	T^{-1}	$k = \pi \frac{L}{U} \nu$
ϱ	Density of displaced water	ML^{-3}	—
σ	Coefficient of rate of change of heading term in proportional rudder control	T	$\sigma' = \frac{U}{L} \sigma$
φ	Phase shift of transfer function	—	—
χ	Heading angle from fixed datum direction	—	—
ψ	Angle of yaw, or heading error	—	—
ω	Angular frequency of ship yawing oscillations	T^{-1}	$k = \frac{L\omega}{2U}$, $\omega' = \frac{L}{U}\omega$
ω'	Non-dimensional frequency of ship yawing oscillations (*cf.* k)	—	(cf. above)
ω_n'	Non-dimensional natural frequency of undamped free oscillations	—	—
ω_1	Angular frequency of rudder oscillations	T^{-1}	$k_1 = \frac{c\,\omega_1}{2U_1}$

The subscript a applied to a symbol indicates an amplitude value, and the subscript c a constant value in steady circling. In Section 9 the subscript c also refers to "closed loop", whereas the subscripts e, i and o indicate the error, input and output signals of a servo mechanism.

Other subscripts, *e. g.* v, $\dot{r}$, β, are applied to define the partial derivatives of the hydrodynamic forces and moments with respect to these modes of motion.

A $\cdot$ (dot) over a symbol stands for a derivation with respect to time.

A $'$ (prime) of a symbol is used to indicate the non-dimensional form (*cf.* p. 38.)

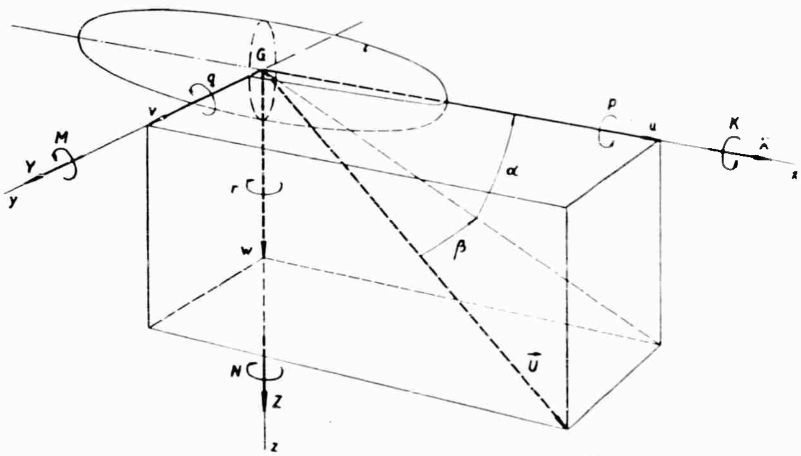


Fig. 1. Body axes (xyz) and flow axis $(\vec{U})$.

3. Course Stability and Manoeuvring of Ships — a Brief Outline of Literature

This paper is chiefly concerned with the development of some simplified formulae and criteria in the well-known theory of ship motion, and with a discussion of a few questions related to the parameters of these equations. It is advisable first to give a brief outline of the literature pertaining to such problems; among the titles reviewed here, those by SCHOENHERR, THIEME and BERNDT as well as the recent survey by ST. DENIS and CRAVEN all contain valueable bibliographies.

The kinematics of manoeuvring was gradually developed by many authors, and mention must here be made of the treatise by POLLARD and DUDEBOUT (1894), as well as of the paper by ROTHE (1910), in which the virtual inertias were substituted for the inertias of the hull proper. (The concept of virtual mass and virtual moment of inertia had been introduced by GREEN and was already applied by WILLIAM FROUDE in his work on the rolling of ships.) More widely known are the classic memoirs by HOVGAARD (1912), KLEIN (1923) and KUCHARSKI (1932), the latter using the analogy between the hull and a wing. It was clearly shown that the motion of the ship is largely governed by the hydrodynamic forces acting upon the hull itself, and that the function of the rudder is more or less that of a servo controlling the attitude of the hull. In the simple example of a steady turn, for instance, the small effective rudder turning force (including the interference force experienced by the hull) balances the inward lateral hull force and the inertia forces due to radial acceleration. Most rudders of the same area and in roughly similar positions will carry almost the same maximum lift at some optimum helm angle; thus the minimum turning circle is essentially a function of hull form parameters.

In spite of these early findings, the majority of model tests on manoeuvring was devoted to the study of rudder stock moments and of rudder forces or "initial ship turning moments". This fact might have been a consequence of the performance of the steering engine being the only manoeuvring quality defined in engineering terms and regulated by the classification societies. However, a modification of the rudder and tail fin usually offers the only way of improving the steering characteristics of an existing ship, and often this remedy has proved to be sufficient. Regarding the turning ability of the ship, there has seldom been a serious demand for radical improvements, the adoption of twin spade rudders on destroyers forming a possible exception.

If steering or course-keeping was thus of more importance than minimum turning circles, at least in the eyes of the practising naval architect, stability and control in flight was a still more urgent problem for the aircraft designer at the beginning of the century. The stability of a dynamical system, described by EULER's equations for moving body axes, had already been studied by ROUTH (1884), who linearized the external forces for small deviations from a state of equilibrium and pointed out the significance of the coefficients of the determinantal equation. By introducing the force-velocity derivatives and, when dealing with lighter-than-air bodies, acceleration derivatives in the linearized expressions for the aerodynamic forces BRYAN (1911), BAIRSTOW (1913), JONES (1921) and others established a theory for the complete motion in the six degrees of freedom. (The acceleration derivatives, of course, are virtual inertias in a real fluid with signs reversed.) The modern presentation of the normalized equations is due to GLAUERT (1927). Admittedly the theory for a long time remained of academic interest chiefly, as it was confined to flying with controls fixed or free and there was no means of analysing the human pilot.

WEINBLUM and KÜNZEL made use of these methods in two papers on ship motion published before the last world war (1937, 1938), with an actual example of a ship model. In a contemporary paper CONTENSOU (1938) demonstrated how the stability of a stationary motion could be inferred from an implicit representation of the simultaneous component equations in semi-polar coordinates. These three authors also gave the analytical criterion of an inherent dynamical stability on straight course now in use; it requires a knowledge of stationary derivatives only.

The conditions of directional stability in automatic steering along a fixed course had been discussed by MINORSKY (1922) by means of the technique of added derivatives, applied to a simplified one-degree-of-freedom oscillation and including several types of position and rate control. Although he was able to form some conclusions as to time lag effects too, it was not until after the war that this analysis was renewed. Until then, the development of automatic steering devices had also been left more or less without the aid of naval architects.

During the war years the demand for automatic controls and high manoeuvrability of submarines and surface ships made necessary an intensified experimental research in this field, and the data then accumulated indicated new aspects of the problems. In his 1944 essay DAVIDSON demonstrated what could and could not be achieved by pure common sense reasoning applied to such experimental figures.

In 1946 DAVIDSON and SCHIFF took a large step towards a better understanding of the interrelation between the performance of a ship on a straight course and in turning, pointing out the nonlinearities in the behaviour of the unstable ship and including an interesting treatment of the transients when entering a steady turn. They also used linear theory for establishing a formula for the radius of the turning circle.

ABKOWITZ started his investigations on the control of submarines from the general equations of motion set up for aerial flight. Most of his results were given in unpublished lecture notes, although his approach forms the basis for the consistent nomenclature suggested by the Society of Naval Architects and Marine Engineers (1950). Reference is also given to a paper by ABKOWITZ (1957) on ship motion in pitch and heave, where the same general equations are used.

The choice of an adequate nomenclature is of utmost importance to a fruitful communication of ideas, and realising the increasing complexities of problems of the future, ST. DENIS and CRAVEN (1958) recently put forth a simplified matrix notation. It is believed that the SNAME 1950 code will still be a standard for use in most practical applications, however.

The works of DAVIDSON and SCHIFF, and of ABKOWITZ, made much to stimulate other authors. From the Stevens school originate, among many others, a report by WILLIAMS (1948) on initial stage motion and a paper by SCHIFF and GIMPRICH (1953), who studied an automatic control system, where the rudder angle called for is proportional to a combination of heading deviation and rate of change of heading, and which has a behaviour with a close resemblance to the automatic pilots used in practice. Two types of time lag were investigated, and it was shown to which extent such lags could be tolerated in the steering of ships in a calm sea. PETERS (1948) made a formal investigation of the motion and stability of submarines, also using the virtual inertias in the development of the equations, and perhaps the main differences of the two schools are to be found herein. In the discussion of a paper on the stability of a towed ship by STRANDHAGEN, SCHOENHERR and KOBAYASHI (1950), WEINBLUM and CONTENSOU gave an indication of the precautions necessary when applying the virtual inertias in the equations. This question will be considered later.

The envelope of the hydrodynamic reactions on a towed ship, drifting with zero angular velocity, is known as "la courbe dérive"

and the position of its "centre initial" or its point of reflection on the ship centerline being aft of the point of towline attachment is one of the conditions necessary for the stability of the towed ship. For the self propelled ship, DIEUDONNÉ (1949) introduced the corresponding envelope of the resulting forces in turning with zero drift, the apex of which he showed must be aft of the centre of gravity or forward of "le centre de dérive initial" for stable conditions to prevail.

Within the last ten years many important contributions have added to the knowledge of special problems. Independent of earlier papers HORN (1951) derived similar conditions for the dynamical stability in a calm sea, and he also investigated the response to disturbances and the relative importance of some of the force derivatives using the hydrofoil boat as an illustrative analytical example. DAVIDSON (1948) and GRIM (1951) studied the loss of stability experienced by a ship in following seas. For the study of steering in a regular seaway, as the first step to a still more general approach, the ideas of WEINBLUM will probably have a fundamental bearing.

Following the paper by WEINBLUM and ST. DENIS (1950) on the motion of ships in a seaway WEINBLUM (1951) applied the results for the transverse force and yawing moment due to buoyancy effects in the seaway to determine approximate formulae for the hull reaction derivatives with respect to a small change of wave train heading. Although the virtual inertias and rotary derivatives were assumed to remain largely unaffected by the waves, and although a uniform speed of advance was still accepted, the problem of directional stability in waves was shown to lead to considerable mathematical difficulties.

In a recently published paper, RYDILL (1959) avoided some of these difficulties by a further approximation of the expression of exciting forces in a long-crested sea with small wave amplitudes, when heading error could also be kept small in automatic steering. From an analysis of the steering in a regular sea with a high frequency of encounter, he demonstrated the need for a suitable control system with a low-pass filter to reduce the high frequency movement of the rudder, which he proved could not effectively react against yawing in oncoming seas. The fact that ship response to rudder movement is appreciable at relatively low frequencies only, severely limits the value of forced oscillation by rudder in the calm water of a short towing tank as a method of determining stability characteristics, as will be touched upon later. RYDILL also applied his linear treatment to some studies of steering in an irregular sea.

There is an extensive literature on turning and course-keeping due to Italian, Russian and Japanese authors, which is often less well known or inaccessible to readers in western countries. Reference to the works of CASTAGNETO (1948), CHANOVIC (1948), BASIN (1949) and MOTORA (1946, 1953) will make the list of titles on the kinematics somewhat more representative. Other papers will be referred to below.

Most of the papers so far reviewed have been devoted to the study of the stability of motion or to the calculation of trajectories for a transient or steady motion, assuming that the forces on the hull and rudder, or their derivatives, are known. Having established a relation between these forces and the resulting motion, it is clear that it will be possible, in special cases at least, to reverse the procedure, *e. g.*, to deduce the forces acting on the hull from a knowledge of the forcing rudder function and the motion observed for the ship. Such methods were first applied to manually piloted aircrafts, by NORTON (1923), GARNER (1924, 1926) and others, but more recently, in forms of frequency response tests, they have become powerful tools in the analysis of automatically steered bodies.

Actually the first theoretical study of automatic feed-back control systems, by MINORSKY (1922), was concerned with the steered ship, although L. B. SPERRY had successfully flown a gyrostabilized flying boat eight years earlier. The auto pilot, the steering engine and the ship with its rudder all form different components of a closed loop system, each component characterized by its transfer function or the complex ratio of output to input. The theory for such control systems and their stability has been developed in electric network engineering, and it is natural that the dynamic problem lends itself to studies in analog computers, where each component is represented by its equivalent electric circuit. The stability of the closed loop system may be judged from the total open loop response recorded at several frequencies, without a knowledge of the individual transfer functions. If these individual functions are desired, they may be derived from frequency response tests with each component in open loop, as indicated in the previous paragraph.

Starting from a linearized set of simultaneous equations for the motion of a stable ship-and-rudder system, the differential equation relating, for instance, the yaw angle output to a harmonic rudder function is easily found. The solution of such an equation is made up of a transient term, decaying by time and being the natural response to any sudden disturbance, and of a forced perma-

nent oscillation with the frequency of the rudder. After some time, only this second term remains, and the transfer function on base of varying frequencies is obtained from a series of observations of amplitude ratios and phase angles. The transfer function may also be found from an analysis of the transient motion following a certain rudder movement, in which case this may be expressed by a sum of sinusoidal terms of different amplitude and frequency; the transient will contain terms of those same frequencies, for which the transfer function is given by the amplitude ratios along the imaginary axis.

Milliken (1947, 1951) and Bollay (1951) have given excellent reviews of the work in this field of aircraft design, and the methods have been introduced in ship and submarine design by Schiff (1948) and Geisberg (1950). There are also a number of classified reports on submarine testing. Running frequency response tests in a model basin with limited length offers special problems, as it may be necessary to include long periods corresponding to the natural oscillation of a model of a submarine with metacentric stability or of a surface ship in a regular sea, in which a minimum of rudder motion is desired. Some of these problems have been investigated by Berndt (1956), in whose treatise are discussed the relative merits of harmonic or transient responses in submarine and model testing.

Of special interest are the possibilities for such analysis offered by the standard zig-zag manoeuvring test for surface ships, first suggested by Kempf (1935) for comparative performance evaluation only. In a later paper Kempf (1944) published the results of a large number of such tests, stating common values for the normalized period of a complete manoeuvre, the meaning of which is discussed in terms of the time lags involved in a report by Gimprich and Jacobs (1948).

Nomoto, Taguchi, Honda and Hirano (1957) have made a theoretical and experimental study of the ship response in different steering processes and demonstrated a simplified analysis of the trapezoidal zig-zag manoeuvre by using transfer functions for a suggested first-order equation of motion. Further work along these lines would be of utmost value.

The transfer function, the magnitude of which thus may be derived from experimental observations, is built up of the stability derivatives forming the constant coefficients of the equations of motion. If some of these derivatives can be estimated with acceptable accuracy, by theory or from simple model tests, it may be possible to eliminate the other from the transfer function.

As has already been pointed out, the linear theory gives a linear formula for the steady state angular velocity on base of effective rudder turning moments, which is valid for gentle turns at least. The coefficient determining the slope of this line also contains the stability derivatives, and if some of them, or their relationships, are known, the others may be determined from ordinary full scale trials. Further, if such trial results are available for one ship and if the effect of say a modified stern arrangement will be of importance to the rotary derivatives only, the turning ability of the new design may be predicted with some confidence.

A considerable number of turning circle test results for destroyers and other naval ships have been published by HOVGAARD (1912), PITRE (1934, 1935) and COLE (1938). In this connection SCHOENHERR's analysis (1939) of HOVGAARDS figures should be specially mentioned. Also, in the files of various shipyards, there are numerous steering trial results for ordinary merchant ships, but in most of these cases the turning circles have only been run with the rudder hard over. Of course, such results are not available for an analysis using linear theory, and it is hoped that more information will be obtained from future trials in accordance with a new proposal for steering and turning tests, the latter run with 35° and 20° rudder.

Throughout this review the need for theoretical and experimental research on the hydrodynamic forces on the hull has been strongly emphasized, and many of the modern papers cited also include important contributions in this respect. It is, however, noted that the basic problem had for a long time been ignored by most model experimentors, although submarine models had been subjected to routine testing in three-component stationary balances. The first experiments to support the development of manoeuvring theory and known to the present author were those by CASTAGNETO (1935).

Similar measurements were made by KÜNZEL and WEINBLUM (1938) on two models, one designed with a bulbous bow. Systematic tests to study the influence of hull form parameters were carried out by SUTHERLAND (1948); see also below. Other tests on single models are due to THIEME (1956) and to HORN and WALINSKI (1958), in both these cases to illustrate theoretical investigations on the motion of the ship. Of special interest are some tests with a semi-submerged prolate spheroid described by GAWN (1951). The two last references also report on force measurements on the oscillated models.

In order to simulate the flow conditions due to steady path curva-

ture, cambered models were frequently used in airship model testing, and the limitations of this technique have been discussed by GOURJIENKO (1934). In their 1938 paper KÜNZEL and WEINBLUM studied the stability of a ship model in steady turning by means of force measurements on a cambered model towed along a straight course with different combinations of drift and rudder angle; further tests were made by KEMPF (1945) and more recently by THIEME (1956) and MÖCKEL who compared the results with rotating arm measurements in an extensive investigation of hull forces on a motor launch in permanent motions.

Rotating arm facilities are now found at many model basins, but up till now few results of general interest have been reported. However, some tendencies were furnished by DAVIDSON and SCHIFF (1946) as a basis for the discussion already referred to, and SUTHERLAND (1948) published valuable information from measurements on a series of captive surface models, the hull proportions and deadwood area of which were systematically varied. The models were also towed on a straight course and the results were given in charts of non-dimensional coefficients containing an empirical factor for hull form parameters.

For use with linear theory, model experiments should provide values of force and moment derivatives. As the relative magnitude of the turning radius is limited by the free length of the rotating arm and the length of the smallest model consistent with reliable measurements, it is necessary to extrapolate the results to an infinite radius of curvature. Moreover, the derivatives evaluated from static balances on towing carriages or rotating arms are at best a kind of quasi-static derivatives. In contrast to a body in an ideal fluid, the ships and models are known to experience a transverse force or lift, which in part has its origin in a circulation over the after body. The forces on the body will therefore depend on the history of the motion too, the significant parameter being the reduced frequency or STROUHAL number, demonstrated by DUNCAN and COLLAR (1932) in the case of aerofoils in unsteady motion. In as much as each state of motion may be identified with a series of harmonic components, it is desirable to run the models in a number of sinusoidal motions of different frequencies. The forced oscillation technique, first introduced in wind tunnel testing by SIMMONS (1921), can be adopted to the testing of captive ship and submarine models. The hydrodynamic reactions due to accelerated motions, or the virtual inertias, are also derived from these tests.

In early testing arrangements, the exciting forces were applied through an elastic system; due to the high damping of the model the natural oscillations of that system soon died away and the steady oscillation of the model was then of the frequency of the excitation, lagging behind it by a certain amount. For the reduction of different derivatives it was necessary to record the amplitude and phase lag of the oscillation, and in some cases also the forces applied to the elastic system. In most modern oscillating devices, the models are rigidly attached to the exciting mechanism, whereas the forces acting between model and support are directly recorded by means of small-displacement transducers. In this way it is possible to let the model oscillate in any desired mode of motion. In the so called planar motion mechanism, designed by GERTLER and GOODMAN (1958), submerged models may be run in a pure pitching or heaving motion with respect to the flow axis; the damping and inertia forces, being out of phase and in phase with the motion respectively, are automatically separated in the data processing. For "Schlängelschwingungen" with surface ship models HORN and WALINSKI (1958) made use of a similar arrangement, in which the drift angle was always kept small, although a true yawing oscillation with respect to the flow axis could not be obtained.

With the aim of facilitating a systematic presentation of experimental hull force data and, ultimately, to devise acceptable methods for theoretical predictions, the analogy between a ship hull and a lifting wing has long been advocated. BRARD (1951) and INOUE (1956) both formulated expressions for the transverse hull force from considerations of the vortex system of a turning hull. SILER (1949) and THIEME (1954) applied the results of small aspect ratio lifting line theories with special emphasis on the non-linear effects. FEDYAEVSKY and SOBOLEV (1957) have introduced the theory of JONES (1946) for the non-circulatory lift of slender wings, the use of which may be based on the assumption of a rearward region of vorticity modifying the hull form.

For boat-tailed bodies of revolution, the wing analogy concepts have so far rendered no results. The ideal theoretical distribution of pressure over the surface of an airship or torpedo is known to break down at its extreme tail, giving rise to a resistance to axial motion and to a stabilizing lift at small angles of attack. HARRINGTON (1935) also demonstrated the existence of a pair of trailing vortices originating from the leeward side of an elongated spheroid. The flow pattern

found by HARRINGTON has no resemblance with the flow over the wing model of very small aspect ratio studied by GOLUBEV (1952). PODE is reported to have proved the failure of the wing theory approach to the lift of bodies of revolution and to have laid the foundation of a new theory for such bodies.

4. The Equations of Motion

The general motion of a ship, on the surface or submerged, is that of a rigid body subject to gravity and buoyancy forces, to controlling forces and hydrodynamic reactions and to hydrodynamic or other disturbances or excitations.

In the case of a truly floating ship, the weight of which is just supported by buoyancy due to hydrostatic pressure, gravity exerts a metacentric couple only. The controlling forces are usually effected by a deflection of the rudder surface or by a change of speed of the propeller. This study does not consider manoeuvres involving use of the propeller. The hydrodynamic reactions include the propelling thrust and rudder forces in steady forward motion; these reaction forces depend on the (changing) form of the ship and on the motion and its history.

The hydrodynamic disturbance forces may be due to disturbance velocities, such as current, sea and wind, or they may be caused by the interaction of other ships or foreign boundaries. The first type of disturbance force depends on the relative motion and orientation of the ship, while the second type depends on the position as well.

Here the motion of the ship in response to rudder excitation is assumed to be governed by metacentric and hydrodynamic reactions only, although the effect of external disturbances may be present in the initial conditions. The rudder deflection itself may be controlled by the heading error sensed by the helmsman or an automatic pilot.

The mass distribution of the ship and its velocities $\vec{U} = u\hat{i} + v\hat{j} + w\hat{k}$ and $\vec{\Omega} = p\hat{i} + q\hat{j} + r\hat{k}$, referred to a set of moving body axes, define the linear and angular momentum vectors $m\vec{U}$ and $\vec{H}$ in that same system $(Gxyz)$, the origin of which is taken to be the centre of gravity of the ship. In particular, if the axes are chosen to be the principal axes of the body, the angular momentum is given by $\vec{H} = I_{xx}p\hat{i} +$ $+ I_{yy}q\hat{j} + I_{zz}r\hat{k}$.

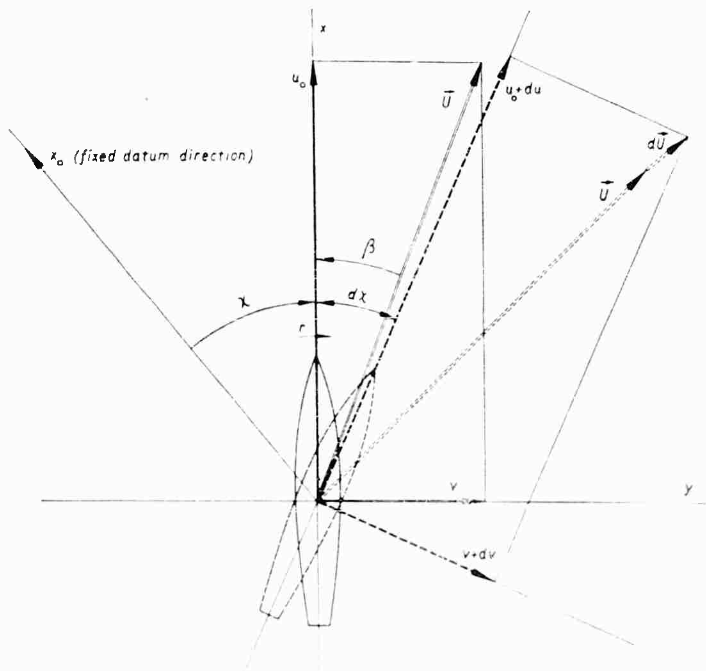


Fig. 2. Motion in the horizontal plane.

Within a small interval of time, the changes of these momentum vectors with respect to an inertial system of co-ordinates fixed in space are seen to be unaffected by a constant translational velocity and only dependent on the infinitesimal partial changes with time and angular attitude, so that the total rate of change is given by

$$\left.\begin{aligned} \frac{d}{dt}(m\vec{U}) &= m\dot{\vec{U}} + \vec{\Omega} \times (m\vec{U}) = \vec{F} \\ \frac{d}{dt}(\vec{H}) &= \dot{\vec{H}} + \vec{\Omega} \times \vec{H} = \vec{M} \end{aligned}\right\} \qquad (4.1)$$

The dots denote the time rate of change with respect to body axes.

Resolving the equations of motion into separate equations for each single component of the forces acting gives the well known EULER equations to be found in most text books on classical mechanics.

$$\left.\begin{aligned}
m(\dot{u} - rv + qw) &= X + X(\delta) \\
m(\dot{v} - pw + ru) &= Y + Y(\delta) \\
m(\dot{w} - qu + pv) &= Z + Z(\delta) \\
I_{xx}\dot{p} + (I_{zz} - I_{yy})qr &= K + K(\delta) + K_g \\
I_{yy}\dot{q} + (I_{xx} - I_{zz})rp &= M + M(\delta) + M_g \\
I_{zz}\dot{r} + (I_{yy} - I_{xx})pq &= N + N(\delta) + N_g
\end{aligned}\right\} \qquad (4.2)$$

Here the terms of δ refer to a rudder excitation while K_g, M_g and N_g are the components of the metacentric couple.

In general, the components of the right hand members will include the effects of added inertias as well as damping, restoring and exciting forces, which are functions of the true motion in all six degrees of freedom. For many practical purposes, however, it is possible to decouple one or more of the equations and to separate the component motions.

In that which follows, this paper will only deal with the stability and performance of a surface ship controlled in a permanent motion along a preset heading on the compass, or run on a straight course or in a steady turn with fixed rudder. Before proceeding to a further simplification of the equations, however, it is necessary to agree upon a suitable representation of hull reactions and controlling forces.

5. The Representation of Hull Forces

In the study of directional (controlled) stability on course, or the inherent stability in straight or curved motions, interest centres around the small deviations, or oscillations, of the body axes with respect to their mean positions. It may then be seen that, to a first order of magnitude, the hydrodynamic reaction along the longitudinal axis and the forward speed will both be unaffected, and consequently the equation of freedom in surge will be decoupled. (It is generally accepted that a turning ship takes up a steady state motion within the second quadrant of the turn, which is certainly true for a high-powered and highly stable ship like a destroyer, but which is far from the case for a heavy tanker with a moderate dynamic stability.) In the light of practical experience, the coupling between motions in the

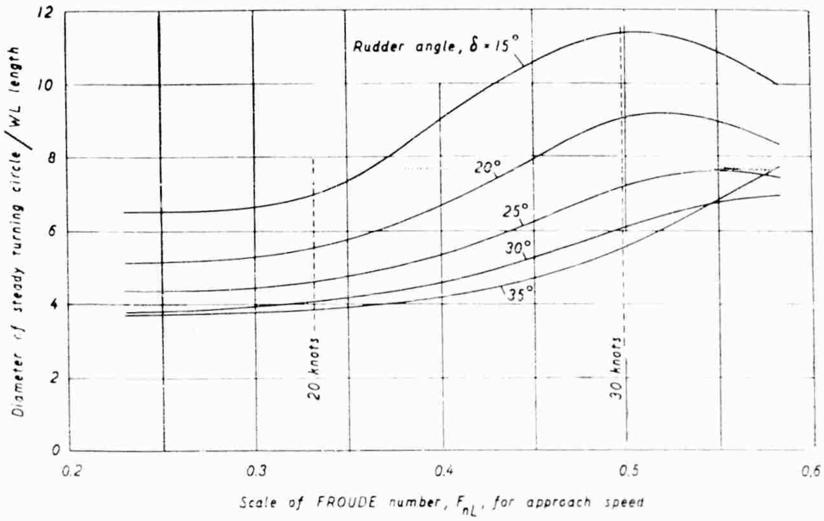


Fig. 3. Influence of speed on turning performance. Curves for British I-class destroyer, compiled from data published by COLE 1938.

different planes may also be ignored. Thus, rolling oscillations are assumed to be small in the permanent motions studied, whereas the mean list of the turning ship may define a new geometry of its hull.

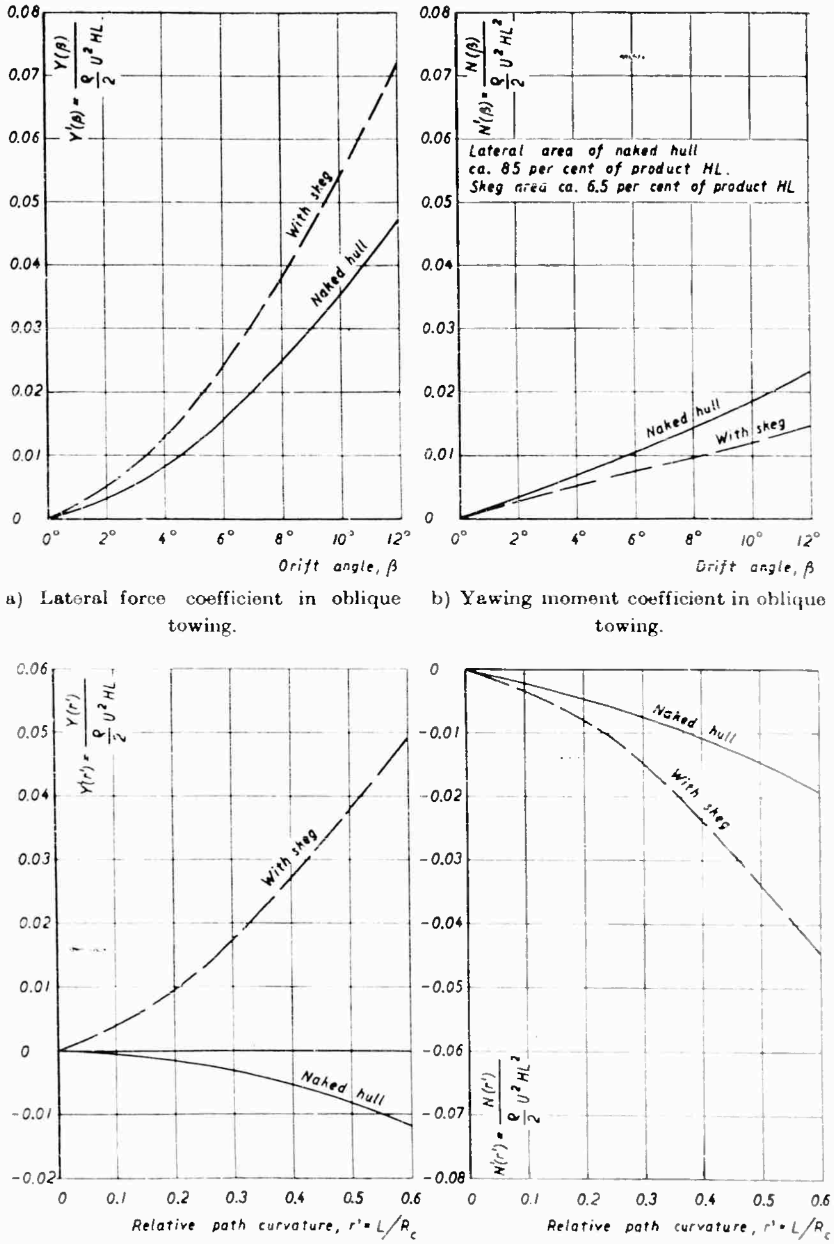
For a surface ship running at FROUDE numbers exceeding $F_{nL} = U/\sqrt{gL} = 0.3$, say a destroyer at a speed of more than 20 knots, wave formation is generally found to cause a change of trim and stability characteristics, and it is necessary to investigate the motion at each different speed. For lower FROUDE numbers, the motion will be assumed not to depend on actual speed, although such a dependence has been anticipated by ST. DENIS from trials with a naval auxiliary with zero rudder. (See discussion of 1946 paper by DAVIDSON and SCHIFF). When dealing with the trajectories and stability of curved motion, cross-flow parameters may have some bearing in this respect, and on base of tests with yawed models THIEME (1956) suggests the use of a draught FROUDE number so that $F_{nL} < 0.7/\sqrt{L/H}$; however, that figure will probably vary with the type of the ship. For the destroyer type the dominating phenomenon seems to be the marked trim and sinkage in the wave hollow at higher speed-length ratios, and the reader may be familiar with the sight of water piling up against the outer quarter in a turn. Fig. 3 shows the effect of speed on the turning radius of an I-class destroyer as reported by COLE (1938).

Without entering into the details of hull force calculation, it may be desirable to facilitate the discussion by use of the results of the systematic model tests due to SUTHERLAND (1948). The curves in Fig. 4 are deduced for a hull with the proportions of a destroyer, showing the static lateral force and yawing moment coefficients on a base of drift angle in straight towing and relative path curvature in rotating arm tests with zero drifting velocity. (Note that the angle of drift and the relative path curvature express the small transverse and angular velocities in a non-dimensional form; *cf.* Section 7). In all cases there is a pronounced nonlinearity, mainly caused by the increasing effect of viscous cross-flow resistance at the higher local drift angles. As a matter of fact, most other tests do show a somewhat wider range of linear dependence.

For small deviations from the steady state motion of a dynamical system it is generally assumed that the restoring and damping forces and moments can be represented by means of partial "stability" derivatives with respect to the changes of positions and velocities of the system components. To some extent it is also possible to express the equality of forces in a new steady state by these derivatives.

As has already been stated, the hydrodynamic forces acting on a body in a real fluid will be functions of the complete motion, *i.e.* they will depend not only on the velocities but also on their rates of change with time and also their time integrals. In the equations of motion the acceleration derivatives may always be associated with the inertias of the body to form the virtual inertias; they are more or less free from any effects of viscosity, and may be calculated as for the ideal case. In the presence of a free surface, the hydrodynamic inertias will depend on the boundary conditions and therefore on the nature of the motion. It will here be assumed that the vertical velocities of the surface will be small so that gravity forces may be neglected when compared to the inertia forces. In that case, the free surface boundary conditions remain the same for all types of horizontal motion. A similar assumption has already been introduced for the damping forces in the non-wavemaking range of forward speed. The damping forces, however, are directly or indirectly due to viscosity, and the vortices momentarily left behind the ship still influence the flow around it; strictly, then, there can be no constant derivatives in a non-stationary motion.

For the underwater portion of a naked hull, the "deeply submerged" values of the coefficients of accession to inertia (k_y, k_{zz}) may be



a) Lateral force coefficient in oblique towing.

b) Yawing moment coefficient in oblique towing.

c) Lateral force coefficient in circling with zero drift.

d) Yawing moment coefficient in circling with zero drift.

Fig. 4. Static lateral force and yawing moment coefficients. Influence of skeg on naked hull. Example calculated for destroyer form ($L/B=10$, $B/H=3$) from model test data published by SUTHERLAND 1948.

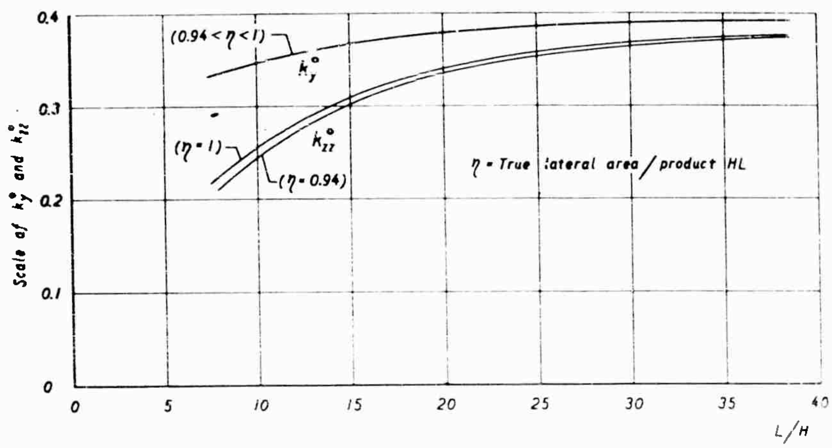


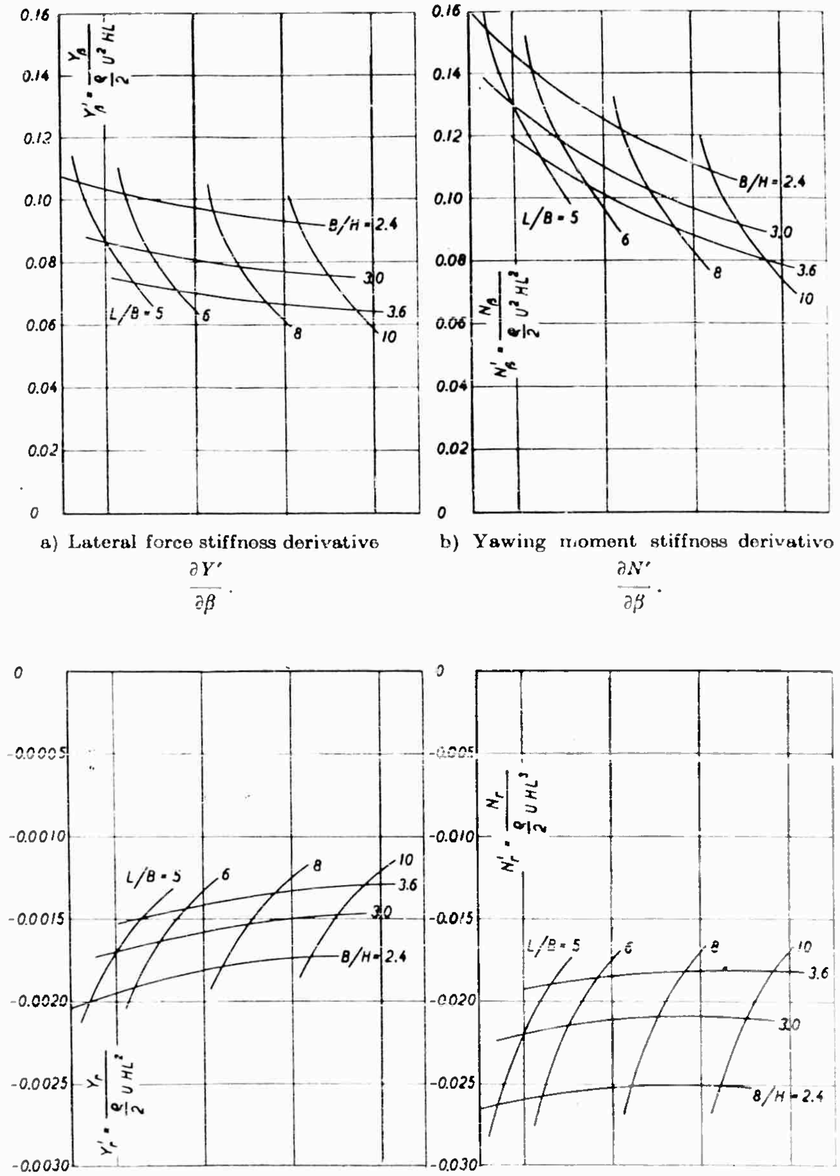
Fig. 5. Coefficients of accession to inertia corrected for free surface effect neglecting gravity

approximated by LAMB's (1918) values for the equivalent ellipsoid of the mirrored body. It is readily seen that the presence of a free surface will lower these values. For the problems considered it appears appropriate to use the reduction factor found by LOCKWOOD TAYLOR (1930) for an elliptical cylinder floating half submerged. Then approximately, with accepted superscript,

$$\left.\begin{aligned} k_y^0 &= 0.4\ k_y \\ k_{zz}^0 &= 0.4\ k_{zz} \end{aligned}\right\} \tag{5.1}$$

Curves of k_y^0 and k_{zz}^0 are shown in Fig. 5. The equivalent ellipsoid has been defined as that one which has the length and lateral area of the mirrored under-water hull. To find the hydrodynamic mass and moment of inertia, the two coefficients must be multiplied by the mass and moment of inertia of the displaced water respectively.

Once more using SUTHERLANDS data, the influence of hull proportions on the stationary force and moment derivatives has been calculated and presented in the form of carpet diagrams in Fig. 6. Of course the linear approach is no longer justified whenever the squares and products of the deviation velocities may not be ignored. Some further consideration of this question will be given in Section 10. It has been noted that the damping derivatives are functions of the velocities and their history. In a transient motion with no violence,



a) Lateral force stiffness derivative $\frac{\partial Y'}{\partial \beta}$.

b) Yawing moment stiffness derivative $\frac{\partial N'}{\partial \beta}$.

c) Lateral force rotary derivative $\frac{\partial Y'}{\partial r'}$.

d) Yawing moment rotary derivative $\frac{\partial N'}{\partial r'}$.

Fig. 6. Static lateral force and yawing moment derivatives of naked hulls. Influence of hull proportions. Calculated from model test data published by SUTHERLAND 1948.

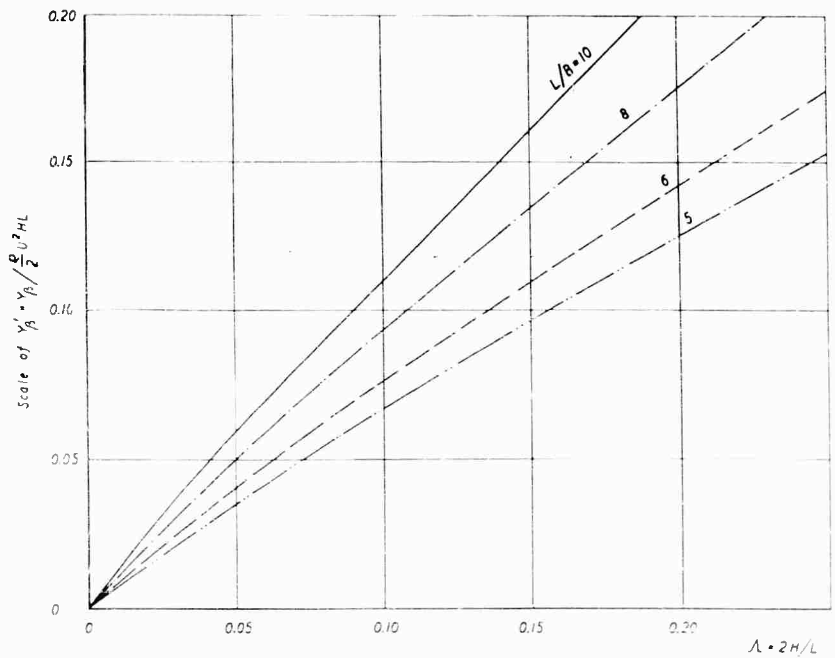


Fig. 7. Naked hull lateral force derivative Y'_β on a base of lateral area aspect ratio $\Lambda = 2\ H/L$. Calculated from model test data published by SUTHERLAND 1948.

the history of that motion is soon forgotten, and it is then appropriate to make use of the derivatives measured or calculated for stationary conditions. When so used, these derivatives may be called quasi-static derivatives.

Moving obliquely in an ideal fluid the boat-tailed hull would experience no transverse force or "lift" but only a free broaching moment. (MUNK, 1924). From model tests with torpedo shaped bodies as well as with ship hulls, this static moment is known to be reduced by 20 to 35 per cent due to the stabilizing effect of the rearward viscous lift. In the case of normal hull forms with not too gentle bilges, where cross-flow stream pattern development is more or less spontaneous, the transverse lateral force may be predicted by use of low aspect ratio wing analogy. For small angles of drift, that is small drifting velocities, this force is then presumed to be mainly non-circulatory in nature, although the viscous separation on the after body is its primary origin, causing a finite width of the trailing edge. In Fig. 7 the lateral force derivatives for each L/B is represented by

an almost straight line to the base of $2\,H/L$, confirming results analogous to the additional mass theory by R. T. JONES (1946), by which the lift of a slender wing is proportional to aspect ratio and angle of attack.

Of course the existing theory of lift for low-aspect ratio wings in unsteady motion, corresponding to the MUNK-JONES theory of steady flight, does not include any dependence of wake figuration. As noted, however, the hull-wing analogy assumes a viscous wake, and in the important cases of harmonic motions with frequency ν the derivatives will depend on the values of the non-dimensional frequency parameter, or reduced frequency, $k = \pi \dfrac{L}{U} \nu$, just as has been shown for aerofoils in pitching and normal translations. (The notation k has been adopted by most authors on the aerodynamics of non-stationary motion. In this report it is used parallel with the alternative form $\omega' = \dfrac{L}{U} \omega = 2\pi \dfrac{L}{U} \nu = 2\,k$, more common in the study of the motion of rigid bodies.) For sufficiently slow oscillations these "exponential derivatives" are practically equal to the quasistatic ones, and the relative waviness of the wake behind the ship may be taken as a measure of the departure from quasi-static conditions.

SZEBEHELY and NIEDERER (1953) studied a prolate spheroid of fineness ratio 7 in free oscillations at various values of reduced frequencies in the range $1.2 < k < 8$; some of the results are given in Fig. 8. The moment derivatives with respect to angular and transverse velocities are both found to decrease with increasing k, the decrease from static values being 5 per cent at k equal to 0.7—1.0 approximately. Some tests on ship models by HORN and WALINSKI (1959) indicate a more significant departure of force and moment derivatives from static values at frequencies between 0.6 and 1.2, but as variation of frequency parameters was not originally intended, these tests are less conclusive with regard to actual frequency dependence.

It is well known that a good helmsman does not try to check the yawing oscillations of his ship in a short oncoming sea; as will be seen in Section 9 the ship response to that rapid rudder motion would then be very small. Similary, if a normal seagoing ship is forced to yaw with the frequency of its rudder continually moved 10° to either side, *i. e.* with a reduced frequency for the hull of the order of $k = 5$

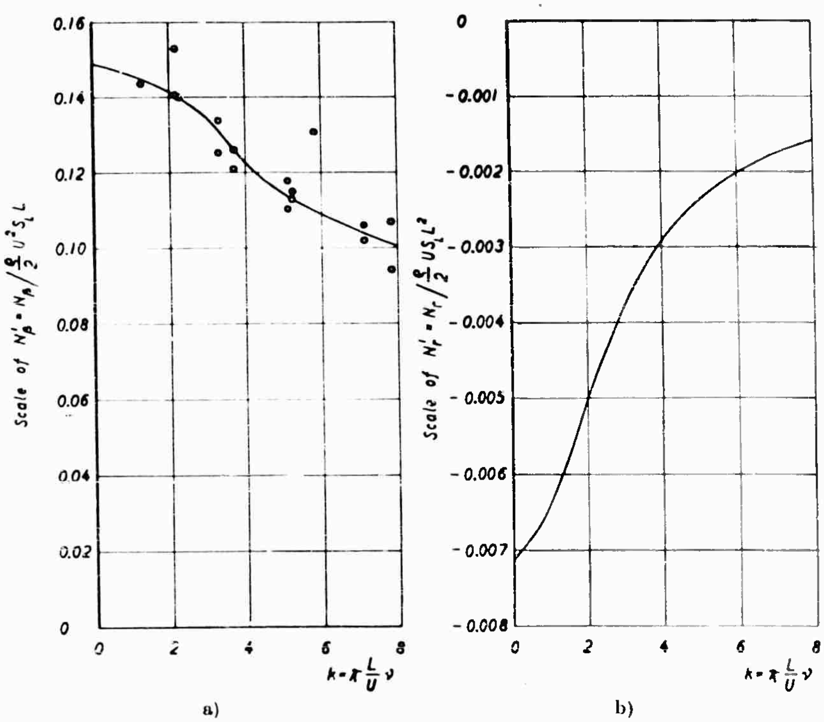


Fig. 8. Unsteady effects on moment stiffness and rotary derivatives of a prolate spheriod of fineness ratio 7. Recalculated from model test data published by SZEBEHELY and NIEDERER 1953.

or $\omega' = 10$, it will perform only moderate oscillations, which are of little value for further analysis. On the other hand, for a ship in a standard zig-zag (KEMPF) manoeuvre involving time lags the normalized period of the almost sinusoidal oscillation is known to be of the order of $\frac{TU}{L} = 8$, corresponding to $k \approx 0.4$, and the use of quasistatic derivatives in the calculations is not likely to introduce any serious errors. Moreover, if such a manoeuvre is made the subject of frequency response analysis, the "transfer functions" are inherently dependent on the frequency. Nevertheless there is need for further oscillator tests with captive models in the low as well as in the high frequency region. Attention must also be paid to the effect of boundary layer development in unsteady motions.

6. The Forcing Rudder Function

The rudder serves the two-fold function of stabilizing a straight motion by fin effect and controlling the ship in steering and manoeuvring. The forces acting on the rudder in its zero position will be included in the hull forces and the forcing rudder function will be due entirely to the deflection of the movable part of the rudder. Owing to the mutual interaction between hull and rudder, parts of the controlling forces are carried by the hull itself. Propeller influence is manifested by the higher velocity of the flow around a rudder in behind position and by the fin effect of the propeller in a non-axial stream. (*Cf.* Section 10.)

There is a vast literature on the theory and practice of rudder design, a review of which is beyond the scope of this paper. A few notes pertaining to rudder motion are, however, given here.

For the rudder, or for an aerofoil in general, the lift may be calculated from a knowledge of the quasi-steady distribution of vorticity over the surface, of the momentary "starting" vortices newly shed into the wake and of the changes of momentum due to the additional hydrodynamic mass. The effect of a varying angle of attack becomes apparent already at the quite slow motions of conventional rudders. In the initial stage there will be a small lift chiefly due to additional mass, whereas the development of lift due to circulation will lag behind the motion. It is clear that not only the rudder stock moment but also the lift force will depend on the position of the stock. In Fig. 9 is shown the result of calculations made by Okada (1958) for the lift of a two-dimensional wing or freerunning rudder with 30 per cent area balance, deflected to 35° at a normal rate; no account was taken of the stalling of the wing. When tested in a high speed wind tunnel the smooth aerofoil may attain a maximum section lift coefficient of say 1.4, but for a ship's rudder the maximum lift coefficient in stationary deflection usually does not exceed 1.0, due to the increased loss of boundary layer energy behind surface roughnesses. When the rudder is rapidly moved beyond the angle of stationary stall, a somewhat higher lift may be momentarily built up, but this will be of no importance in the steering process.

If the rudder is oscillating with constant frequency of sufficient magnitude, the stationary value of lift amplitude is not realised but the lift will be a function of the frequency parameter, as already

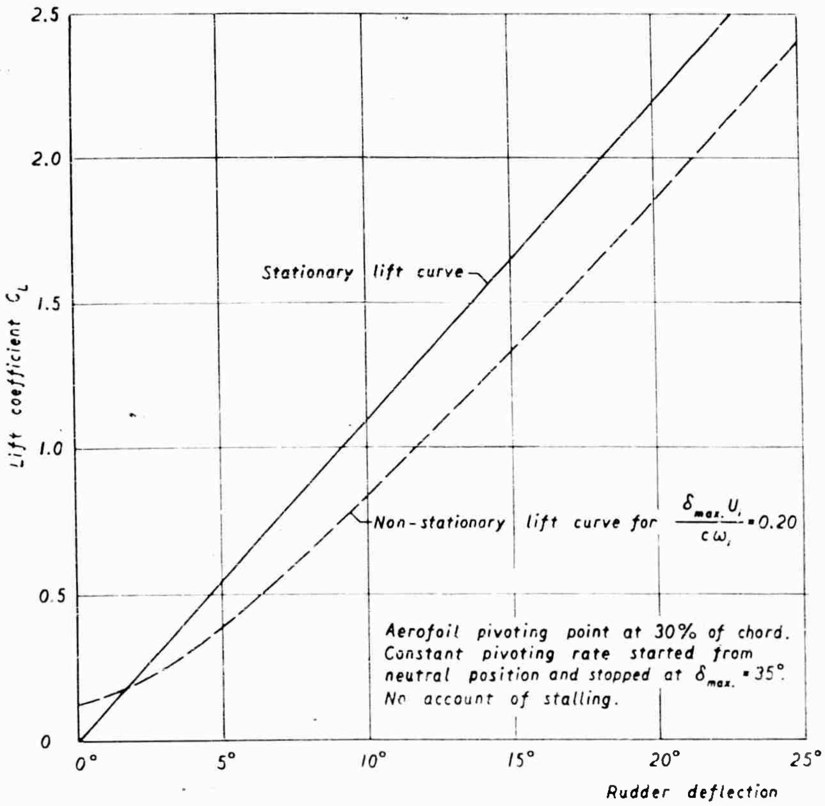


Fig. 9. Effect of rudder motion on two-dimensional section lift coefficient. Figure due to OKADA 1958.

stated. In practice, this may be the case for a ship-rudder system with automatic compass control steering.

The sinusoidal motion $\delta = \delta_a e^{j\omega_1 t}$ of an aerofoil (a rudder) about an axis at distance a from the leading edge may be separated into a transverse velocity equal to

$$\bar{w} = \bar{w}_a e^{j\omega_1 t} = \left[U_1 \delta_a + j \left(\frac{c}{2} - a \right) \delta_a \omega_1 \right] e^{j\omega_1 t} \tag{6.1}$$

all over the chord and a rotation about the mid-chord point, giving rise to a transverse velocity

$$\Omega_1 x = \Omega_{1a} x e^{j\omega_1 t} = j \omega_1 \delta_a x e^{j\omega_1 t} \tag{6.2}$$

at distance x aft of that point. The lift of an aerofoil element of unit span, expressed by VON KÁRMÁN and SEARS (1938) for the two pure oscillations considered, may be written

$$L(\bar{w}) = [L(\bar{w})]_a e^{j\omega_1 t} = \left[\pi \varrho c U_1 \bar{w}_a C(k_1) + \frac{\pi}{4} \varrho c^2 \dot{\bar{w}}_a\right] e^{j\omega_1 t} \quad (6.3)$$

$$L(\Omega_1) = [L(\Omega_1)]_a e^{j\omega_1 t} = \left[\frac{\pi}{4} \varrho c^2 U_1 \Omega C(k_1)\right] e^{j\omega_1 t} \quad (6.4)$$

The second term inside the bracket of equation (6.3) is due to the apparent mass effect, which will of course have no bearing on the ideal lift of the purely rotating wing. $C(k_1)$ is a complex function of the reduced frequency of the aerofoil, $k_1 = \frac{c\,\omega_1}{2\,U_1}$, known as the THEODORSEN function, and is reproduced in Fig. 10. Realising that $\pi \varrho c U_1 \bar{w}_a$ is the stationary lift of an element of the rudder, deflected to $\delta_a = \frac{\bar{w}_a}{U_1}$, and combining equations (6.1)—(6.4), it is convenient to write for the controlling force of an oscillating rudder

$$Y(\delta) = \Phi(k_1)\, Y(\delta_a)\, e^{ifkt} \quad (6.5)$$

where

$$\Phi(k_1) = C(k_1) + j\,\frac{k_1}{2}\left\{1 + \frac{C(k_1)}{1 + j\left(1 - \frac{2a}{c}\right)k_1}\right\} \quad (6.6)$$

$$f = \frac{c/L}{U_1/U} \quad (6.7)$$

The function $\Phi(k_1)$ (for $a/c = 0.25$) is also shown in the diagram of Fig. 10. The real component of the position vector, from the origin to a point on the curve corresponding to the frequency indicated, expresses the ratio of lift amplitude to steady state value, whereas the argument of the vector is the phase angle by which the force leads the motion. At frequencies likely to occur in the normal steering operation, the force will lag slightly behind the rudder deflection.

Another constant lag in every steering process; manual or automatic,

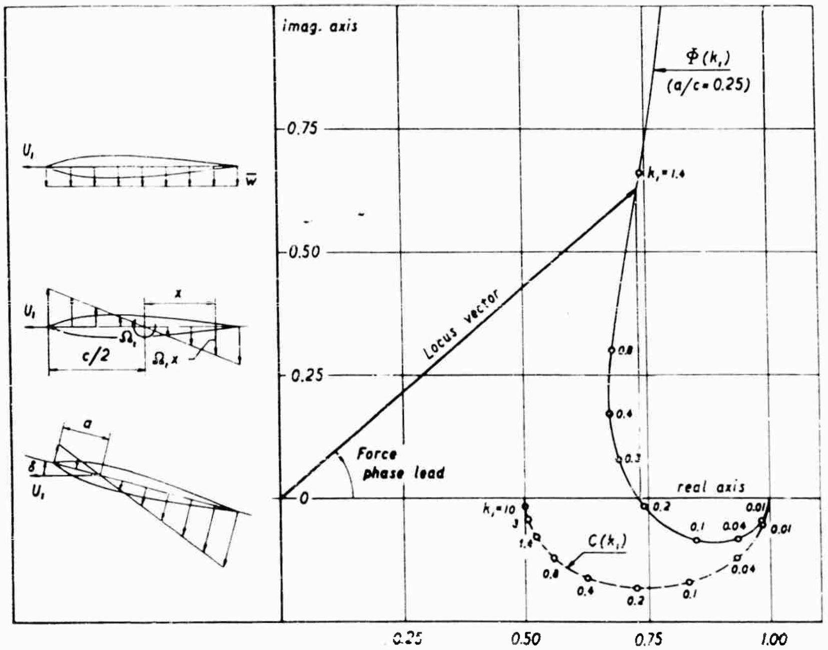


Fig. 10. The THEODORSEN function $C(k_1)$ and the function $\Phi(k_1)$ for the lift of an oscillating aerofoil. $Y(\delta,\ k_1) = \Phi(k_1)\ Y(\delta_a)\ e^{jfkt},\ f = \dfrac{c/L}{U_1/U}$. The real component of the locus vector equals the ratio of the oscillating lift amplitude to its steady state value. A positive argument of the vector is the phase angle by which the force leads the motion.

will be due to the finite time necessary to convert the error signal detected into an output signal to the steering engine; both these components of a constant time lag will be small compared to the delays introduced by the finite rate of the steering gear, however.

Most modern steering engines are designed to move the rudder with an essentially constant speed, the rudder turning at that speed as long as a control signal is transmitted to the steering engine. A gyro pilot may be used to switch on this signal, calling for a correcting rudder at a certain small deviation from the desired heading, whereas a contact on the rudder may stop the motion at a suitable angle; the rudder remains in this position until the ship has swung over to the other side of its course and the rudder is then reversed. (The zig-zag test proposed by KEMPF makes use of a similar scheme with larger

amplitudes). Controls like this simple on-off type will often be found on torpedoes and amateur model yachts, where the rapid oscillations of hull and rudder are less harmful. More often a follow-up mechanism assures a "proportional control" of the rudder. Due to the finite rudder speed, these systems may be self-exciting, however, and excessive heading oscillations may be built up.

In order to overcome these difficulties the automatic pilot must be made to anticipate the motion of the ship, much in the way an experienced helmsman gives an auxiliary rudder. In practice this is accomplished by means of a feed back of rudder motion to the heading error detector, as in many commercial applications, sometimes also by adding some kind of rate of change of heading control, similar to the pitch rate component of submarine depth control systems. In effect, both these methods correspond to a "proportional plus rate control"; the first one often incorporates a non-linear character, as will be denoted in Section 9.

The ship response to the trapezoidal periodic steering in a calm sea is an almost harmonic oscillation in yaw and sway, in as much that the inertia of the ship filters out the effects of the high frequency components of the rudder motion. The qualitative discussion of the behaviour of automatically steered ships is notably facilitated if the motion of the rudder is also represented by a continous analytical function.

As has been demonstrated by SCHIFF and GIMPRICH (1949) and as will be seen in the next section, there are some advantages in expressing the rudder lag in terms of a differential equation. Let it be accepted that the rate of turning the rudder is gradually diminished to make the rudder come neatly into the new position, in such a way that the speed is roughly proportional to the deflection remaining, *i.e.* to the difference between the instantaneous position of the rudder and the position called for, δ^*; then $\dot{\delta} = \frac{1}{\bar{t}}(\delta^* - \delta)$, or

$$\delta + \bar{t}\,\dot{\delta} = \delta^* \tag{6.8}$$

In normal steering in a calm sea the rudder is "flipping" with amplitude δ^* on either side of a mean position, and its initial rate may be taken to be equal to the mean rate recorded at a hard-to hard manoeuvre, $\dot{\delta}_0$, of the order of 2.5° per second. The time lag coefficient $\bar{t} = \frac{2\,\delta^*}{\dot{\delta}_0}$ is seen to vary with the amplitude and may be determined

by an iterative process if necessary. In good weather δ^* is say 2.5° only, and then $\bar{t} \approx 2$ seconds. When using the normalized equations of motion derived in the next section the appropriate time lag coefficient will also be non-dimensional, being $\bar{t}' = \frac{U}{L}\bar{t}$, with a magnitude of the order of 0.1 for a modern seagoing ship.

The differential equation of the rudder control considered now becomes

$$\delta' + \bar{t}'\dot{\delta}' = \gamma'\psi' + \sigma'\dot{\psi}' \tag{6.9}$$

where the heading error is set equal to the angle of yaw, *i. e.* the angle measured from a fixed direction on the compass (the x_0-axis of the space system) to the direction headed by the ship. If the ship is to starboard of the course (positive ψ) this will call for a port rudder (positive δ^* in the sense of a positive rotation about the z-axis.)

7. The Simplified Equations

The general equations of motion have been formulated in Section 4 and the validity of a series of simplifying assumptions in the treatment of surface ship steering has been discussed in Sections 5 and 6. The main assumptions are once more listed here:

The weight of the ship is supported by buoyancy and the ship has adequate transverse stability. The rolling motion is negligibly small. A constant heel may be considered as defining a new hull form.

The wave-making and its effects are moderate and the trim of the ship and the form of its underwater hull are not sensitive to small changes of speed.

The ship operates in calm deep water far from other ships or foreign boundaries.

The motion of the ship may be regarded as taking place in the horizontal plane only. The forward speed is largely unaffected by the yaw and sway of the ship and these motions are unaffected by small changes of that speed. The equation for surge may therefore be decoupled.

The ship is steered by one or more stern rudders along a preset heading or left to move on a straight course or in a permanent circle with fixed rudder. The manoeuvres studied involve only small changes of the motion, thus justifying a linear approach.

The forces on hull and rudder may be separated into hull reactions and active rudder forces.

The hull reactions, experienced by the hull and rudder in zero position, depend on the instantaneous motion of the ship, described by its velocities and accelerations in the horizontal plane and by the frequency of that motion; if the motion is not periodic, the changes are supposed to take place so slowly that the forces may be independent of the history of these changes.

The hull reactions are expressed by means of velocity and acceleration derivatives.

In steady oscillation, the values of the velocity derivatives may depart from the quasi-steady values measured in oblique towing or rotating arm tests.

Some of the acceleration derivatives may be calculated by use of the coefficients of accession to inertia of ellipsoids half submerged in an ideal fluid, neglecting gravity effects, but there is also a lateral force due to yawing acceleration and a yawing moment due to the acceleration in sway.

The active rudder forces, carried by the rudder and its hull image, depend on the deflection of the rudder relative to the ship. If the rudder is oscillating, the forcing rudder function may also depend on the frequency of that oscillation.

The deflection of the rudder may be constant or it may be controlled by the motion of the ship. When the deflection is moderate, the rudder forces will be proportional to the deflection; in other cases the rudder force itself may be introduced as a variable in the equations.

When automatic steering is considered, the controlling force will be approximated by a continous function of the yawing of the ship, the rudder position being proportional to a combination of compass heading error and rate of change of heading as sensed a few seconds earlier. The time lag will be due chiefly to finite rudder speed, and it is assumed that it may be expressed by what is known as an exponential lag.

With the approximations accepted and with signs and symbols as given is Section 2, the equations governing the motion of the ship and rudder will simply read

$$\left.\begin{aligned} &(m - Y_{\dot{v}})\,\dot{v} - Y_v\,v - Y_{\dot{r}}\,\dot{r} - (Y_r - mu_0)\,r = Y_\delta\,\delta \\ &N_{\dot{v}}\,\dot{v} + N_v\,v - (I_{zz} - N_{\dot{r}})\,\dot{r} + N_r\,r = \lambda\,L\,Y_\delta\,\delta \\ &\delta + \bar{t}\,\dot{\delta} = \gamma\,\psi + \sigma\,\dot{\psi} \end{aligned}\right\} \qquad (7.1)$$

Here $Y_\delta \delta$ is the total lateral force, on hull and rudder, due to the deflection of the rudder from its zero position, and λL is the distance between the point of application of this force and the centre of gravity of the ship.

Two of the coefficients of the linear and angular accelerations of equations (7.1) are seen to be virtual inertias, made up of the mass elements of the ship and the water entrained in the motion, whereas the others, $Y_{\dot{r}}$ and $N_{\dot{v}}$, are hydrodynamic derivatives experienced in real fluids only and usually rather small. It is also noted that the mass, appearing in the term associated with the derivative Y_r, does not include any added inertia, a fact which may be worth some attention.

Fig. 2 shows a ship moving in the horizontal plane, this motion being represented by the velocity vector $\vec{U}$, the velocity of yaw, r, and the momentary heading χ, being the angle from some fixed datum direction to the longitudinal axis of the ship. In the next instant, the small changes $d\vec{U}$ and $d\chi$ will appear, associated with a change of the transverse momentum with respect to space axes, and within the short interval considered, the virtual mass momentum along the transverse axis of the ship is to a first order increased by $m_y\, dv + m u_0\, d\chi$. Note that the hydrodynamic mass $m_x - m$ does not enter this expression. Now, if the time rate of change of this momentum is equated to a transverse force component $Y = Y_v\, v + Y_r\, r + Y_\delta\, \delta$ the y-equation will be

$$m_y\, \dot{v} - Y_v\, v - (Y_r - m u_0)\, r = Y_\delta\, \delta \qquad (7.2)$$

in agreement with the corresponding equation in (7.1), except for the acceleration derivative $Y_{\dot{r}}$ now neglected.

When the motion of the ship is expressed by its translatory and rotatory velocities, the partial derivatives of the forces and moments with respect to these velocities may all be considered as damping derivatives, or they may be termed resistance derivatives as in the original literature of aircraft stability. However, the equations will now be modified to forms in which they will be more suitable for nautical applications.

Assuming a constant forward speed $u_0 \approx U$, it is convenient to replace the change of drifting velocity by a corresponding change of angle of drift or sideslip, defined by $v = - U\,\beta$, thereby also intro-

ducing the angular displacement or stiffness derivatives anticipated in Section 5.

The angular velocity r of the ship relative to the z-axis through its centre of gravity is by definition equal to the rate of change of heading or heading error as soon as this axis is parallel to the vertical, which has here been postulated; thus $r = \dot{\psi}$ and $\dot{r} = \ddot{\psi}$.

As was seen the hull and rudder will usually be subject to hydrodynamic reactions, which are non-linear functions of the drift angle and path curvature. When treating the dynamical stability of a ship in a turn, where the mean values of β and r are not small, the derivatives must correspond to the slopes of these functions at the points of circling equilibrium. (In those cases the linear approach is of course not adequate for estimating the radius of the turning circle as a function of the rudder angle, which will be further discussed in Section 10.) The change of the total rudder force due to the disturbances of that equilibrium is assumed to be approximately equal to the change of that same force in the case of an undeflected rudder, which change is included in the hull force differential.

The drift angle and rate of change of heading may be determined from careful plots of compass readings and double bearings to fixed objects, or by use of modern radio-navigating aids if available. The hydrodynamic derivatives will be kept in the form most consistent with captive model test procedures.

The equations of set (7.1) then become

$$\left.\begin{aligned}
&(m - Y_{\dot{v}})\, U\, \dot{\beta} + Y_{\beta}\, \beta + Y_{\dot{r}}\, \ddot{\psi} + (Y_r - m\, U)\, \dot{\psi} = -\, Y_{\delta}\, \delta \\
&N_{\dot{v}}\, U\, \dot{\beta} - N_{\beta}\, \beta + (I_{zz} - N_{\dot{r}})\, \ddot{\psi} - N_r\, \dot{\psi} = -\, \lambda\, L\, Y_{\delta}\, \delta \\
&\delta + \tau\, \dot{\delta} = \gamma\, \psi + \sigma\, \dot{\psi}
\end{aligned}\right\} \qquad (7.3)$$

In order to make the equations more suitable for a qualitative discussion, they will now be normalized by the use of non-dimensional coefficients, denoted by a prime. Following Glauert (1927), unit mass will be written $\mu \dfrac{\varrho}{2} SL$, where μ is the relative density, being $\mu = 1$ for the displacement surface ship; thus the mass is $m = m' \dfrac{\varrho}{2} SL$, with the characteristic length equal to the WL length of the hull. If the reference area S is chosen to be the area $\dfrac{2V}{L}$, the non-dimensional

mass is seen to be $m' = 1$, which gives a very simple form to the equations. For the purpose of this paper, however, the reference area for lateral motion, S_L, will be taken as equal to the product of WL length and draught of underwater area. By this convention the non-dimensional coefficients will be related to a "wing" area, while the mass will still appear as m' in all results deduced from the equations of motion. Numerically the value of m' will be $2\,\delta_B \frac{B}{L}$. (The product HL is preferred to the true contour lateral area, as the amount of deadwood or cut-away-area is one of the major parameters affecting the values of some of the coefficients.) Finally, if unit time is the time required to travel one ship length, a time interval t is expressed by the number of ship lengths travelled in that interval. $t' = \frac{U}{L}\, t$; from this are derived the units for linear velocity, U, for angular velocity, U/L, for linear acceleration, U^2/L, and for angular acceleration, U^2/L^2.

A non-dimensional force-angular velocity derivative is defined as the partial derivative of the normalized force coefficient with respect to the normalized angular velocity, and so for instance $Y_r = \partial \left(Y' \frac{\varrho}{2} U^2 S_L\right) \Big/ \partial \left(r' \frac{U}{L}\right) = Y'_r \frac{\varrho}{2} U S_L L$. The diagrams in Fig. 6 exemplifying the discussion of Section 5, have been presented in this form. A complete list of the derivatives and other symbols has been supplied in Section 2.

Now there results

$$\left.\begin{aligned}
&(m' - Y'_{\dot{r}})\, \dot{\beta}' + Y'_{\beta} \beta' + Y'_{\dot{r}}\, \ddot{\psi}' + (Y'_r - m')\, \dot{\psi}' = - Y'_{\delta}\, \delta' \\
&N'_{\dot{\beta}}\, \dot{\beta}' - N'_{\beta} \beta' + (m' k_L^2 - N'_{\dot{r}})\, \ddot{\psi}' - N'_r\, \dot{\psi}' = - \lambda\, Y'_{\delta}\, \delta' \\
&\sigma'\, \dot{\psi}' + \gamma' \psi' = \delta' + t' \dot{\delta}'
\end{aligned}\right\} \qquad (7.4)$$

When the time lag is ignored, the third equation may be substituted into the two others, and the effect of rudder control then is merely a change of the damping and stiffness characteristics of the ship. In the general case, time lag is quite important.

The forcing rudder input may contain a constant part associated with a turn of radius $R_c = L/r'_c$, which within the validity of linear

treatment may be estimated from (7.4), the drift angle and rate of change of heading being constant in this state of equilibrium.

$$\frac{R_c}{L} = \frac{Y'_\beta N'_r - N'_\beta (Y'_r - m')}{(N'_\beta + \lambda Y'_\beta) Y'_\delta \delta_c} \qquad (7.5)$$

The terms of both members of the equations (7.4) thus in balance may be eliminated before proceeding with a further investigation of the stability of the motion.

In the three homogeneous linear differential equations (7.4), the variables β, ψ and δ are now all functions of the independent time variable t', the number of ship lengths sailed. (Below, the symbols of angular deflections will be written without the primes; on the other hand, the prime must not be omitted from a quantity like $\dot\beta'$, of course.) As $de^{st'}/dt' = se^{st'}$, a set of solutions of the form

$$\left.\begin{aligned} \beta &= \beta_i e^{st'} \\ \psi &= \psi_i e^{st'} \\ \delta &= \delta_i e^{st'} \end{aligned}\right\} \qquad (7.6)$$

is always valid for these homogeneous equations, which may then be converted into a system of algebraic equations by the substitution of this solution, *i. e.* by use of the derivation symbol $s = \frac{d}{dt'}$. The new set of equations is now

$$\left.\begin{aligned} &[(m' - Y'_{\dot\beta}) s + Y'_\beta] \beta_i + [Y'_{\dot r} s^2 + (Y'_r - m') s] \psi_i + Y'_\delta \delta_i = 0 \\ &[N'_{\dot\beta} s - N'_\beta] \beta_i + [(m' k_L^2 - N'_{\dot r}) s^2 - N'_r s] \psi_i + \lambda Y'_\delta \delta_i = 0 \\ &[\sigma' s + \gamma'] \psi_i - [t' s + 1] \delta_i = 0 \end{aligned}\right\} \qquad (7.7)$$

When the rudder angle is not related to the heading of the ship but to a known function of time, the third equation vanishes and the two first ones completely describe the motion of the ship; if the rudder input is periodic, the ship is then forced to oscillate with the frequency of the rudder. From the three equations pertaining to the case of automatic steering it is possible to solve the complex natural frequencies of the closed loop system, $s_i = q_i + j\omega'_i$, as well as the ratios β_i/δ_i and ψ_i/δ_i. If all values of s_i are situated in the negative half of the complex plane, *i. e.* if all values of q_i are negative, all the natural

component oscillations of the system will be damped and decay with time.. As a quantitative measure of the directional stability of the steered ship SCHIFF and GIMPRICH (1949) introduced the parameter q, being to the right in the sequence of q_i along the real axis and dominating the motion for large t'. The actual figure of q will not be further considered.

As the algebraic equations of (7.7) all are homogeneous a non-trivial solution not equal to $\beta_i = \psi_i = \delta_i = 0$ demands the determinant of the system to be zero.

In a more general form the three algebraic equations may be written

$$\left.\begin{aligned}(a_{11}s^2+b_{11}s+c_{11})\beta_i+(a_{12}s^2+b_{12}s+c_{12})\psi_i+(a_{13}s^2+b_{13}s+c_{13})\delta_i=0\\(a_{21}s^2+b_{21}s+c_{21})\beta_i+(a_{22}s^2+b_{22}s+c_{22})\psi_i+(a_{23}s^2+b_{23}s+b_{23})\delta_i=0\\(a_{31}s^2+b_{31}s+c_{31})\beta_i+(a_{32}s^2+b_{32}s+c_{32})\psi_i+(a_{33}s^2+b_{33}s+c_{33})\delta_i=0\end{aligned}\right\} \quad (7.8)$$

and the determinantal equation then is

$$\begin{aligned}|A_{ij}| = A_{31}\,(A_{12}\,A_{23} - A_{13}\,A_{22}) - A_{32}\,(A_{11}\,A_{23} - A_{13}\,A_{21}) + \\ + A_{33}\,(A_{11}\,A_{22} - A_{12}\,A_{21}) = 0 \qquad (7.9)\end{aligned}$$

where $A_{ij} = a_{ij}\,s^2 + b_{ij}\,s + c_{ij}$.

In the determinantal equation of (7.7) $A_{31}, a_{11}, a_{21}, a_{32}, c_{12}, c_{22}, a_{13}, a_{23}, a_{33}, b_{13}$ and b_{23} all are zero, and the equation is given by a quartic in s,

$$D(s) = s^4 + Bs^3 + Cs^2 + Ds + E = 0 \qquad (7.10)$$

If the rudder angle called for were made proportional to rate of change of heading only ($\gamma' = c_{32} = 0$) this equation would have one root equal to zero and there would no longer be a controlled heading. When the rudder remains fixed in a mean position, so that δ_i is constant and equal to zero, the determinantal equation again is reduced to a cubic,

$$\begin{vmatrix}(m' - Y'_{\dot v})\,s + Y'_\beta & Y'_{\dot r}\,s + (Y'_r - m') \\ N'_{\dot v}\,s - N'_\beta & (m'\,k_L^2 - N'_{\dot r})\,s - N'_r\end{vmatrix}\, s = 0 \qquad (7.11)$$

one root of which is seen always to be equal to zero, corresponding to the lack of preference for a defined compass heading. The condition for dynamic stability may then be easily derived from equation (7.11).

8. The Concepts of Stability and the Characteristic Equation

The stability of the permanent motion of an unsteered or automatically steered ship in a calm sea is inferred from the characteristics of the ship response to a small disturbance, *i. e.* of the deviations from the steady state in the time following that disturbance. For a stable motion to persist, these transients will tend to zero as time increases.

A ship is said to be dynamically stable on a straight course or in a turn of constant curvature if, when slightly disturbed from its steady motion, it will soon resume that same motion along a slightly shifted path, without any correcting rudder being applied.

If the ship is dynamically stable on a straight course it is also said to have an inherent lateral stability. If it is not dynamically stable on a straight course with zero rudder, this configuration will usually be dynamically stable in a rather gentle turn in either direction, due to the non-linearities of the hydrodynamic reactions.

A body that has no inherent lateral stability may be made dynamically stable by means of some kind of stabilization by a controlled rudder motion. In practice such a stabilization would rarely be applied to seagoing ships, which instead are made directionally stable by the action of a helmsman or by use of automatic steering devices. A ship may acquire a directional stability without having an inherent lateral stability, which, however, necessitates more severe demands on the time lags tolerated in the steering process.

A steered ship is said to be directionally stable if the rudder can be forced to compensate for repeated disturbances so that the ship will follow a preset course on the compass with only small oscillations in yaw and sway. In a calm sea, the oscillations set up in the system by a finite small disturbance, must tend to zero as time increases.

It is important to note that a dynamically unstable ship with fixed rudder will enter a spiral turn, however small the disturbance met, whereas the stable ship will deviate more from its initial heading the larger the disturbance. In the same way a steered ship will oscillate more the larger the amplitudes of the disturbances.

In the last section, the response of the ship motion in each one of its degrees of freedom was assumed to be made up of a sum of exponential terms, the exponents of which appeared as roots in the characteristic or frequency equation $D(s) = 0$. For the cases considered, this equation was seen to be given by the quartic (7.10), or by an equation of lower degree.

In order to determine whether the dynamical system is stable or not, it is not necessary to evaluate the numerical values of the roots of the characteristic equation but only to find their signs; if the motion is stable, all real roots or real parts of complex roots are negative. The criteria for stability is then made up of the conditions imposed on the constants of a characteristic equation having no positive real roots or positive real parts of complex roots, and the number of such independent conditions must be equal to the number of roots. For shipform bodies, some of these conditions are always satisfied and the criteria for lateral stability may be expressed by a single inequality.

The constants of the characteristic equation are all real numbers, and complex roots therefore will appear in conjugate pairs. If the quartic (7.10) is factorized with respect to its four roots, real or complex, it takes the form

$$s^4 + \left\{-(s_1 + s_2) - (s_3 + s_4)\right\} s^3 + \left\{s_1 s_2 + (s_1 + s_2)(s_3 + s_4) + s_3 s_4\right\} s^2 +$$
$$+ \left\{-(s_1 + s_2) s_3 s_4 - s_1 s_2 (s_3 + s_4)\right\} s + (s_1 s_2)(s_3 s_4) = 0 \qquad (8.1)$$

and it is easily seen that, provided the real parts of all roots are negative, the coefficients B, C, D and E will all be positive. In the way these coefficients are related to each other, this implies three independent and necessary conditions for stability, but there remains to find a fourth one in order to exclude the combinations of positive and negative roots which are also possible.

If all the coefficients are positive, two real roots or the real parts of a pair of conjugate complex roots may still be positive, thus making the system unstable. In the limit, when the real parts of each of these two roots and therefore also the sum of the roots are all equal to zero, the system has just gained a neutral stability. First assume that the imaginary parts are also equal to zero; then D and E will both vanish, and the equation is reduced to one of the second order. The case of complex roots therefore is discriminative. In equation (8.1) let for instance $s_3 + s_4 = 0$, whereas the product of the two roots has a finite positive value. By inspection of equations (7.10) and (8.1) the three unknown $(s_1 + s_2)$, $s_1 s_2$ and $s_3 s_4$ may now be eliminated, and the critical relation of the coefficients is found to be $B_0 C_0 D_0 - B_0^2 E_0 - D_0^2 = 0$, where the subscripts indicate the special values corresponding to this solution. In general, the real term of s_3 and s_4 is not zero and the value of ROUTH's discriminant $R = BCD - EB^2 - D^2$ then also differs from zero; remembering that the coefficients of

the quartic all are assumed to be positive, the sign of the discriminant follows that of the product $(s_1 + s_2)(s_3 + s_4)$, where $s_1 + s_2$ is negative. If $s_3 + s_4$ is not zero but negative, as is required for the stable system, this product is positive.

A dynamical system, the characteristic equation of which is the quartic (7.10), is thus proved to be stable if five quantities all are positive, *i. e.*

$$\left.\begin{aligned} B, C, D, E &> 0 \\ BCD - B^2E - D^2 &> 0 \end{aligned}\right\} \qquad (8.2)$$

By putting one root equal to zero the constant term is $E = 0$ and the criteria for stability of the cubic $s^3 + Bs^2 + Cs + D = 0$ is seen to be that the coefficients all are positive and that $BC > D$. If one more root is zero $D = 0$ and the indication of a stable quadratic is simply that the coefficients of the linear and constant terms both are positive.

In the quadratic determinant of (7.11) the coefficients of s^2 and s are easily proved to be positive, and there will therefore be two negative roots provided that

$$(Y'_r - m')\, N'_\beta - Y'_\beta N'_r > 0 \qquad (8.3)$$

which is the well-known criterion for dynamic stability with fixed controls. (A few comments on this criterion will be added in Section 10.) For normal ships the two roots will be real; the one to the right along the real axis, p_1, becomes more dominating as time increases and it is therefore termed the dynamic stability index. For a stable ship the inverse of $-p_1$ will be roughly equal to the number of ships lengths sailed in the time the magnitude of an initial disturbance will be reduced in the ratio e:1.

The introduction of a new link in the dynamical system — associated with an additional freedom of motion or with a feed back rudder control — materially complicates the form of the determinant. In some cases, however, the characteristic equation may be written down directly from the inspection of a "block diagram", as will be seen below.

When considering the transient motion of a ship in response to an operating rudder, the problem demands not only the solution of the roots of a characteristic equation of higher degree but also the elimination of a set of unknown constants, these being functions of the initial

or boundary conditions of the motion. It is then convenient to make use of operational methods, by which the original differential function of time is transformed into the spectral distribution of a new, algebraic function of the complex frequency variable, and where the initial conditions are introduced at an early stage. Again, for a simple harmonic input, the transfer operator, relating output to input spectral distributions, soon reduces to a steady state transfer ratio in terms of a magnification factor and a phase shift. In the next section the basics of frequency response will be applied to the study of control system stability.

9. Frequency Response and Automatic Control Stability

Let the motion of the rudder be specified by the purely sinusoidal input $\delta = \delta_a e^{j\omega' t'}$, represented by the projection of a rotating vector of magnitude δ_a on to the imaginary axis; the stable ship will then be forced to oscillate in a similar steady mode, so that

$$\left.\begin{aligned} \beta &= R_\beta \delta_a e^{j(\omega' t' + \varphi_\beta)} = R_\beta e^{j\varphi_\beta} \delta \\ \psi &= R_\psi \delta_a e^{j(\omega' t' + \varphi_\psi)} = R_\psi e^{j\varphi_\psi} \delta \end{aligned}\right\} \tag{9.1}$$

where R_β and R_ψ have the meaning of magnification factors (or amplitude ratios) and φ_β and φ_ψ are phase shift angles. Below R and φ without indices will stand for R_ψ and φ_ψ respectively. The complex quantity $\dfrac{\Psi}{\Delta} = Re^{j\varphi}$ is termed the transfer ratio for the frequency considered, and the function $\dfrac{\Psi}{\Delta}(j\omega')$ is the frequency response function, or the transfer function. This solution for the transfer function will be derived in two slightly different ways, the second one perhaps more attractive to the naval architect familiar with the classic studies of rolling in waves.

By the substitution of $j\omega'$ for the complex quantity s in the two equations of (7.7), which characterize the ship-and-rudder system, there result

$$\left.\begin{aligned} &[(m' - Y'_{\dot v}) j\omega' + Y'_\beta] \frac{\beta}{\delta} + [(Y'_r - m') j\omega' + Y'_{\dot r} (j\omega')^2] \frac{\psi}{\delta} = -Y'_\delta \\ &[N'_{\dot v} j\omega' - N'_\beta] \frac{\beta}{\delta} + [(m' k_L^2 - N'_{\dot r}) (j\omega')^2 - N'_r j\omega'] \frac{\psi}{\delta} = -\lambda Y'_\delta \end{aligned}\right\} \tag{9.2}$$

Then the transfer function from rudder deflection to yaw is

$$\frac{\psi}{\Delta}(j\,\omega') = \frac{Y'_\delta}{D\,(j\,\omega')}\,\{[N'_{\dot v} - \lambda\,(m' - Y'_{\dot v})]\,j\,\omega' - (N'_\beta + \lambda\,Y'_\beta)\} \quad (9.3)$$

where the determinant $D\,(j\,\omega')$ has the form of (7.11); it may be factorized as

$$D(j\omega') = [(m' - Y'_{\dot v})\,(m'\,k^2_L - \ldots) \quad N'_{\dot v}\,Y'_{\dot r}]\,(j\omega')\,(j\omega' - p_1)\,(j\omega' - p_2) \quad (9.4)$$

p_1 and p_2 being the real roots of $D\,(s) = 0$ discussed in Section 8. (See also below.) Each factor of the transfer ratio may be regarded as a vector in ARGAND's diagram, $j\,\omega' - p_1$ for instance being that one drawn from the point p_1 on the real axis to ω' on the imaginary axis. By the laws of vectorial multiplication, the amplitude ratio or modulus and the phase shift or argument of $\frac{\psi}{\Delta}\,(j\,\omega')$ are respectively

$$\left.\begin{aligned} R &= \frac{[N'_{\dot v} - \lambda\,(m' - Y'_{\dot v})]\,Y'_\delta}{\omega'^2\,(\omega'^2 + p_1^2)\,(\omega'^2 + p_2^2)}\cdot\left\{\omega'^2 + \left[\frac{N'_\beta + \lambda\,Y'_\beta}{N'_{\dot v} - \lambda\,(m' - Y'_{\dot r})}\right]^2\right\} \\ \varphi &= tan^{-1}\frac{\omega'[N'_{\dot v} - \lambda\,(m' - Y'_{\dot r})}{N'_\beta + \lambda\,Y'_\beta} - tan^{-1}\frac{\omega'}{-p_1} - tan^{-1}\frac{\omega'}{-p_2} - \frac{\pi}{2} \end{aligned}\right\} \quad (9.5)$$

Assuming the acceleration derivatives $Y'_{\dot r}$ and $N'_{\dot v}$ both to be negligibly small, (9.5) takes the form given by SCHIFF (1948); for convenience write $a = -\frac{N'_\beta + \lambda\,Y'_\beta}{\lambda(m' - Y'_{\dot v})}$, whence

$$\left.\begin{aligned} R &= \frac{\lambda\,Y'_\delta}{m'\,k^2_L - N'_{\dot r}}\cdot\frac{\sqrt{\omega'^2 + a^2}}{\omega'\cdot\sqrt{\omega'^2 + p_1^2}\cdot\sqrt{\omega'^2 + p_2^2}} \\ tan\,\varphi &= \frac{\omega'^2\,(p_1 + p_2) - a\,(p_1\,p_2 - \omega'^2)}{a\,\omega'\,(p_1 + p_2) + \omega'\,(p_1\,p_2 - \omega'^2)} \end{aligned}\right\} \quad (9.6)$$

Thus R and φ are functions of the frequency of the forced motion; analytically, moreover, the quantities a, p_1 and p_2 all are formed from the stability derivatives, which themselves may be functions of that motion. (*Cf.* Section 5.) Also note that, at higher frequencies, the amplitude ratio may be somewhat smaller and the true lag between

ship and rudder motions slightly larger due to the unsteady effects of rudder lift indicated in Fig. 10. (See also below.)

Again, alternatively and with the assumptions above, the equations of the forced motion may be written

$$\left.\begin{aligned} (m' - Y'_{\dot{v}})\,\dot{\beta}' + Y'_{\beta}\,\beta + (Y'_r - m')\,r' &= -\,Y'_{\delta}\,\delta_a \sin \omega' t' \\ N'_{\beta}\,\beta - (m'\,k_L^2 - N'_{\dot{r}})\,\dot{r}' + N'_r\,r' &= \lambda\,Y'_{\delta}\,\delta_a \sin \omega' t' \end{aligned}\right\} \qquad (9.7)$$

By elimination of β the oscillation in yaw is determined by

$$\ddot{r}' + 2\,h\dot{r}' + \omega_n'^2\,r' = r'_a\,\omega_n'^2 \left(\frac{\omega'}{a} \cos \omega' t' + \sin \omega' t'\right) \qquad (9.8)$$

where r'_a is the steady turning rate for a permanent rudder deflection equal to the amplitude δ_a, where a has the meaning attached to it above, and where $2\,h = \dfrac{Y'_\beta}{m' - Y'_{\dot{v}}} - \dfrac{N'_r}{m'\,k_L^2 - N'_{\dot{r}}}$. For the inherently stable ship, the coefficients of the inhomogeneous equation have all been shown to be positive, as required for the initial transients to decay; $\omega_n'^2 > 0$ of course leads to the criterion of (8.3). Formally, ω'_n would be the natural frequency of an undamped freely moving ship. Due to the very high damping in yaw, however, $h > \omega'_n$ always, so the real motion of the unsteered ship is of a non-oscillatory character, and there is no resonant frequency of the forced oscillation. Nevertheless, the quotient $A = \dfrac{\omega'}{\omega_n}$, the ideal tuning factor, has a direct bearing on the steady state frequency response.

The steady state particular solution of (9.8) must be of the form

$$r' = r'_a\,(A \cos \omega' t' + B \sin \omega' t') \qquad (9.9)$$

where A and B have to be chosen so as to satisfy the complete equation for all values of ω. Then the magnification factor is $\mu = \sqrt{A^2 + B^2}$, whereas the phase angle, by which the motion of the ship leads the forcing rudder input, is $\varepsilon = tan^{-1} \dfrac{A}{B}$. By integration of this solution for r' the complex transfer ratio of yaw output to rudder input is readily found to be

$$\frac{\Psi}{\Delta} = Re^{j\varphi} = \frac{\mu\,r'_a}{\delta_a\,\omega'}\,e^{j\left(-\frac{1}{\varepsilon}\right)} \qquad (9.10)$$

Solving for R and φ in this expression gives

$$
\left.\begin{aligned}
R &= \frac{\lambda\, Y'_{\delta}}{m'\, k_L^2 - N'_{\dot r}} \cdot \frac{\sqrt{\omega'^2 + a^2}}{\omega' \sqrt{(\omega_n'^2 - \omega'^2)^2 + 4\, h^2\, \omega'^2}} \\
\tan \varphi &= \frac{2\, h\, \omega'^2 + a\, (\omega_n'^2 - \omega'^2)}{2\, ha\, \omega' - \omega'\, (\omega_n'^2 - \omega'^2)}
\end{aligned}\right\} \tag{9.11}
$$

which is easily seen to agree with the previous result.

Perhaps the influence of frequency may best be studied from the magnification factor for rate of yaw amplitude, written in terms of the ideal tuning factor,

$$
\mu = \frac{r'}{r'_a} = \frac{\omega'_n}{a} \cdot \frac{\sqrt{\Lambda^2 + \left(\frac{a}{\omega'_n}\right)^2}}{\sqrt{(1 - \Lambda^2)^2 + 4 \left(\frac{h}{\omega'_n}\right)^2 \Lambda^2}} \tag{9.12}
$$

For $\Lambda = 0$ it is always $\mu = 1$, of course, and for oscillating rudders the rate amplitude decreases with increasing frequency; it is then also a function of ship parameters, condensed in the quantities $\frac{a}{\omega'_n}$ and $\frac{h}{\omega'_n}$. Fig. 11 shows an example of the transfer function from rudder deflection to yaw, using the equations (9.6) for separate plots of amplitude ratio and phase shift for a modern tanker form.

For the numerical calculations the following approximate figures have been estimated:

L/B	B/H	S_R/S_L	λ	δ_B	m'	k_L	N'_r
7.5	2.5	0.015	0.505	0.78	0.208	0.25	−0.095
$N'_{\dot r}$	$N'_{\dot v}$	N'_{β}	Y'_r	$Y'_{\dot r}$	$Y'_{\dot v}$	Y'_{β}	Y'_{δ}
−0.0040	∼0	0.070	0.025	∼0	−0.080	0.23	0.050

By use of the linear formula (7.5), the diameter of a 20° rudder turning circle may be seen to be roughly 5.5 ship lengths, which is a normal value for this type of ship. From the determinant of equation (7.11) the two roots p_1 and p_2 are found to be -0.30 and -6.10 respectively, i. e. the ship has a moderate inherent stability on course.

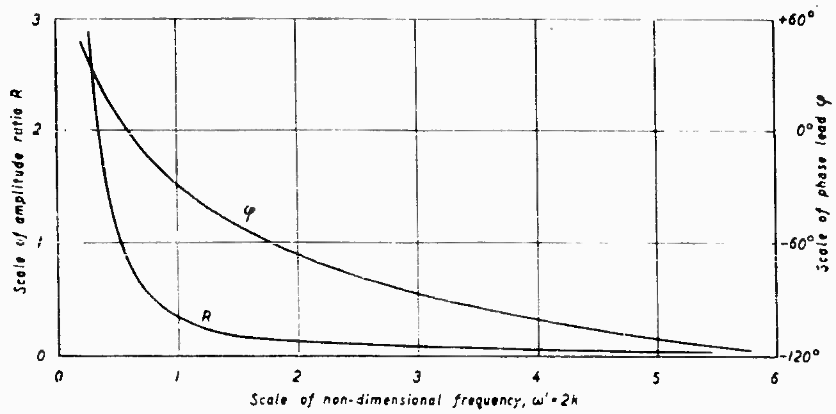
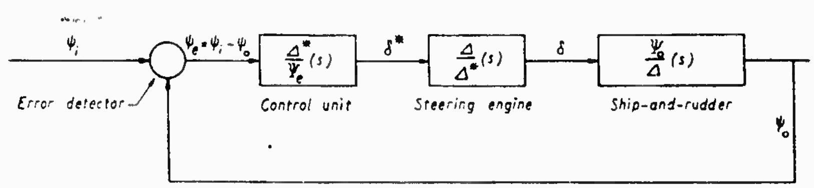


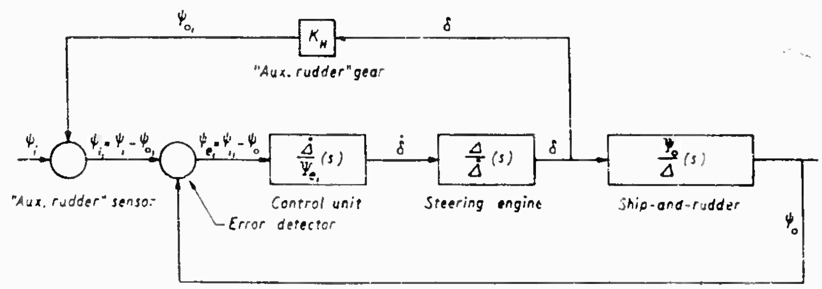
Fig. 11. Transfer function (frequency response function) from rudder deflection to yaw. Example calculated for tanker form from data estimated in the text.

When analyzing the harmonic response of a ship or model run at different rudder frequencies the equations of (9.6), (9.11) or (9.12) may all be utilized to determine the stability index or other hydrodynamic properties, expressed by combinations of the stability derivatives. The techniques involved in such procedures are discussed at some length in the report by SCHIFF already referred to. In this case, the values of the derivatives are assumed to be the same in different runs, which has been seen to be reasonably true only when moderate frequencies are used. Of course the equations may also be used for a calculation of the transfer function from a knowledge of the stability derivatives, preferably derived from captive tests at several frequencies. It is often of interest to obtain a result for the transfer function, whereas the individual parameters are not required; this is the case when considering the automatically steered ship.

The ship-and-rudder then forms part of a closed loop servo system, built up from different blocks or servo amplifiers. For each block — control unit, steering engine *etc.* — the transfer ratio may be calculated be methods analogous to those just used, or it may be evaluated from frequency response tests in the component workshop. The total open loop transfer ratio may be derived from full-scale experiments at sea, where the steering engine receives a signal which is as closely sinusoidal as possible, and where the control unit output is recorded; also, as the transfer ratio product is commutative, the heading output may be



a) System as treated in text.



b) Alternative system with rudder motion feed back in place of rate of change of heading control.

Fig. 12. Block diagram of simple automatic steering systems.

recorded in a test, where the sinusoidal signal from a low frequency generator is fed into the control unit.

When the feed back loop is closed, the output from the ship-and-rudder block is added to the input to the control unit (*n. b.* with signs reversed) and the control output is made the new input to the steering engine. (See Fig. 12 a.) The object of the automatic control is to make the true heading output ψ_0 as near as possible to the desired course on the compass; the momentary heading error $\psi_e = \psi_i - \psi_0$ is sensed by a detector and the signal may be amplified and converted in the control unit to transmit an order for rudder angle δ^*. Due to the steering engine delays, the actual rudder angle response to this order will be given by $\delta + l' \dot{\delta} = \delta^*$, corresponding to a transfer function $\frac{\Delta}{\Delta^*} = \frac{1}{1 + l' j \omega'}$. The non-stationary rudder lift relationship established in Section 6 suggests the introduction of a new block for the analytical representation of the transfer function from rudder angle to rudder force; of course its effect is present in a total open loop transfer function deduced from experiments, but at present it will not be included in the examples calculated below.

As has already been pointed out, there is in many commercial control systems a feed back of the rudder angle output, which by an "auxiliary rudder gear" is translated into a suitable regain of heading. (See alternative block diagram, Fig. 12 b.) In such a system, the controller generally orders a constant rate deflection of the rudder until the "virtual" heading error is reduced to a certain small angle allowed, whereas the reverse rudder manoeuvre does not start until that same small angular error is sensed on the opposite bow. The simple analysis presented below is not readily applied to a non-linear steering control of that kind, however.

The transfer function of the control mechanism here studied will satisfy the equation $\delta^* = \gamma' \psi_e + \sigma' \dot{\psi}'_e$, so that $\dfrac{\Delta^*}{\Psi_e} = \gamma' + \sigma' j\omega'$. The heading output ψ_0 is related to a sinusoidal signal ψ_e fed into the control by the complex open loop transfer ratio

$$\frac{\Psi_0}{\Psi_e} = \frac{\Delta^*}{\Psi_e}\,\frac{\Delta}{\Delta^*}\,\frac{\Psi_0}{\Delta} = \frac{\lambda Y'_\delta}{m' k_L^2 - N'_{\dot{r}}} \cdot \frac{(-a)\,\gamma'}{p_1 p_2} \cdot \frac{\left(1 - \dfrac{j\omega'}{a}\right)\left(1 + \dfrac{\sigma'}{\gamma'} j\omega'\right)}{j\omega'\left(1 - \dfrac{j\omega'}{p_1}\right)\left(1 - \dfrac{j\omega'}{p_2}\right)(1 + t' j\omega')} \tag{9.13}$$

the corresponding over-all amplitude ratio and phase shift being characteristic for each frequency. This response is shown in the open loop polar plot in Fig. 13 a, calculated for the tanker form already used. Four cases are considered, varying in turn the rudder lag coefficient t' and the control parameters γ' and σ'; the full line curve indicates a steering arrangement for which $t' = 0.1$ and $\gamma' = \sigma' = 1$.

The way in which the open loop transfer ratio varies with the frequency offers an important test of the stability of the closed loop system with one-way feed back engaged. A small disturbance of the motion of the ship calls for a correcting rudder, and so an oscillation will be set up. This oscillation may tend to zero as time increases, in which case the system is said to have at least an absolute directional stability; in practice, the satisfactory operation of an automatic pilot even in a "calm" sea does require a certain rate of decay of the "calm sea oscillations", or a certain degree of relative directional stability, providing a safeguard against the effects of "noise" and inaccuracies of

the theory. In this introductory study the criterion of absolute stability will be considered, using the NYQUIST diagram for the single loop servo system. In electronic amplifier design practice, there are some simple rules for the stability margins of "gain" and phase lead which are usually regarded as sufficient, and they have been suggested for guidance in ship control work also. (See below.)

In an expression for the total open loop transfer function written like that in (9.13), the product of all factors not dependent of the frequency is termed the static loop gain K. The dynamic loop gain for each specified frequency is the amplitude ratio of ψ_0 and ψ_e at that frequency.

In Fig. 13 the open loop response curve may be regarded as the locus of the vector quantity $KG(j\omega')$, also being the locus of an output vector from the origin of a diagram, in which the error vector or open loop input is represented by the unit vector along the positive real axis. The magnitude and phase shift of the closed loop response may then be found from a simple addition of vectors as shown in Fig. 14 a.

For the feed back system there is

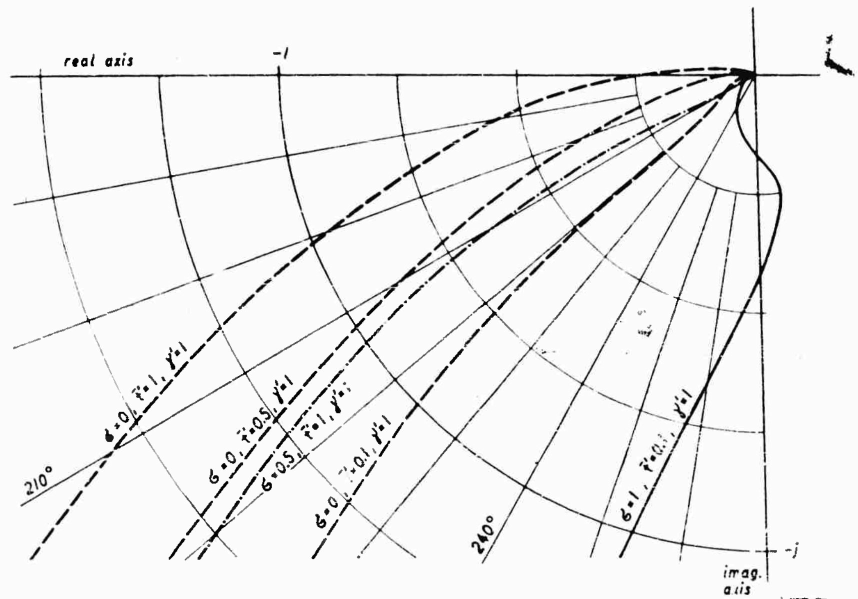
$$\left.\begin{aligned} \psi_0(j\omega') &= KG(j\omega')\cdot\psi_e(j\omega') \\ \psi_e(j\omega') &= \psi_i(j\omega') - \psi_0(j\omega') \end{aligned}\right\} \tag{9.14}$$

from which the closed loop transfer function is given by

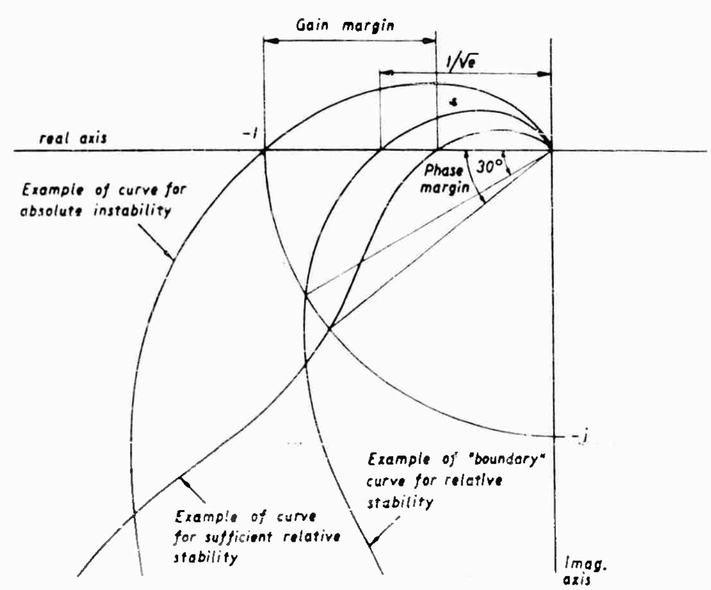
$$\frac{\psi_0}{\psi_i}(j\omega') = \frac{KG(j\omega')}{1 + KG(j\omega')} \tag{9.15}$$

At a certain frequency, given by a point on the open loop output locus, the closed loop magnification factor M is the ratio of the distances from this point to the origin and to the $(-1, 0)$ point respectively.

Suppose the compass input to be disengaged. It is then possible to make the feed back control maintain an oscillation of the ship with constant amplitude at that frequency, for which the open loop phase shift is equal to 180°, provided that the dynamic loop gain can be made equal to unity by a proper selection of static gain, adjustable in terms of control unit parameters. A small compass deviation signal introduced in the error detector is sufficient to cause excessive oscillations,

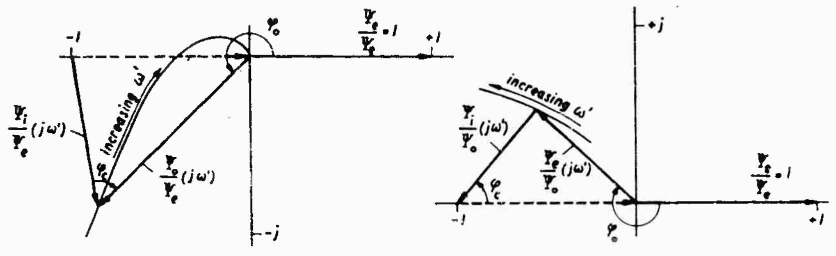


a) Polar plot of open loop frequency response $\frac{\Psi_o}{\Psi_e}(j\,\omega')$. Curves calculated for tanker example analyzed in text.



b) Absolute stability according to the NYQUIST criterion and relative stability as judged from accepted practice.

Fig. 13. Closed loop stability from open loop frequency response.



a) The direct transfer ratio for a given frequency is equal to the ratio of two vectors,
$$\frac{\Psi_o}{\Psi_i} = \frac{\Psi_o}{\Psi_e} : \frac{\Psi_i}{\Psi_e}.$$

b) The inverse transfer ratio is obtained from a simple addition of vectors,
$$\frac{\Psi_i}{\Psi_o} = 1 + \frac{\Psi_e}{\Psi_o}.$$

Fig. 14. System transfer function from open loop frequency response.

however, as according to (9.15) the closed loop transfer function now has a pole at $(-1, 0)$.

For stability to exist in the normal control case, therefore, the frequency response curve must not pass through this point but leave it to the left when viewed in the direction of increasing frequency. The NYQUIST (1932) criterion for the stability of a single loop feed-back system with stable components requires that its open loop transfer function must not encircle the $(-1, 0)$ point. A simple proof of this statement, using the inverse transfer functions, has been given by EVANS (1954).

In (9.15) was given the system transfer function as derived from the open loop frequency response; its general form is obtained by the substitution of the complex quantity s for $j\,\omega'$. The poles of this new function, which determine the character of the transients following a small disturbance, are the zeros of the equation $1 + KG\,(s) = 0$ and correspond to the roots of the characteristic quartic discussed in Section 8. That same expression appears in the numerator of the inverse system transfer function

$$\frac{\Psi_i}{\Psi_0}\,(s) = 1 + \frac{1}{KG\,(s)} \tag{9.16}$$

the construction of which is demonstrated in Fig. 14 b. Note that the direction of positive phase shift is now reversed, and that the magnification factor is simply the reciprocal.

In the case considered it is

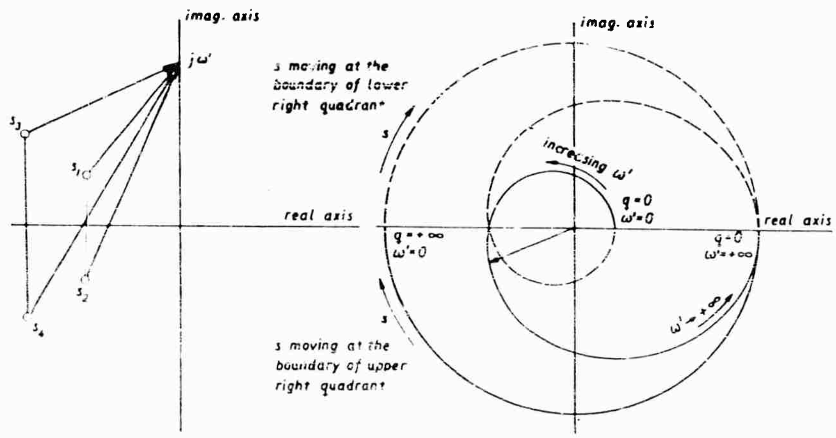
$$\frac{\Psi_i}{\Psi_0}(s) = \frac{s\left(1-\frac{s}{p_1}\right)\left(1-\frac{s}{p_2}\right)(1+l's)+K\left(1-\frac{s}{a}\right)\left(1+\frac{\sigma'}{\gamma'}s\right)}{K\left(1-\frac{s}{a}\right)\left(1+\frac{\sigma'}{\gamma'}s\right)} \tag{9.17}$$

The characteristic equation resulting from (9.17) is identical with (7.10) except for a constant multiplier. If the control unit transfer function is reduced to $\frac{\Delta^*}{\Psi_e}(s) = \sigma' s$, *i. e.* if the rudder angle called for is proportional to rate of change of heading only, one root will always be equal to zero, and the ship will then not be directionally stable. (*Cf.* eq. (9.13) and Section 7.) Of course the conditions for directional stability may now be tested by use of the algebraic rules developed in Section 8. Thus the characteristic equation of the tanker example with $\sigma' = \gamma' = 1$ and $l' = 0.1$ reduces to

$$s^4 + 16.4\,s^3 + 80.7\,s^2 + 52.3\,s + 19.07 = 0 \tag{9.18}$$

the coefficients of which satisfy the ROUTH criterion for stability, (8.2), *i. e.* all the roots (s_1, s_2, s_3, s_4) are in the left half of the complex plane. Here, however, the application of frequency response data will be considered.

In the general case the quartic equation will have two pairs of conjugate complex roots, as is indicated for a stable system in Fig. 15 a. The numerator of (9.17) may be factorized assuming these roots to be known, and for a certain value of s the total expression for $\frac{\Psi_i}{\Psi_0}(s)$ may then be plotted as a product of complex numbers representing vectors drawn to this point s, each from a fixed point in the plane. (This product is a new complex number, the argument of which is the sum of the arguments of the individual factors and which has a magnitude equal to the product of their magnitudes; it may therefore be conveniently deduced by a graphical addition of vectors in an auxiliary diagram having arguments and logarithms of magnitudes along the abscissa and ordinate axes.) Of special interest is the variation of $\frac{\Psi_i}{\Psi_0}(s)$ when s moves up the imaginary axis of Fig. 15 a.



a) Factor vectors from roots of $\frac{\Psi_i}{\Psi_0}(s)$ in the s-plane.

b) Numerator product of vectors in the $\frac{\Psi_i}{\Psi_0}(s)$-plane.

Fig. 15. To illustrate the proof of the NYQUIST criterion.

First consider the complex representation of the numerator alone. For s starting from zero towards increasing values of ω', the corresponding numerator vector moves away from its initial position along the positive real axis, rotating in a positive (anticlockwise) direction as long as s does not leave the imaginary axis. When ω' is very large, the four factor vectors are all parallel with an argument equal to 90°, and the numerator vector then has rotated through 360°, as seen in Fig. 15 b. If s now moves downward along an arc bounding the upper right quadrant of the complex plane, all the factor vectors swing back to 0° and the numerator vector, therefore, also turns back in a negative direction all the way to the positive real axis. When s then procedes around the lower right quadrant, the numerator vector locus repeats a similar track in the mirrored plane, as shown by the dotted curve in the figure. It is thus noted that the vector locus in this case does not enclose the origin as s encircles the right half of the complex plane. If, however, one pair of roots of the quartic had been situated in that right half plane, two of the factor vectors would each have completed a full swing from 0° to 360° and the numerator product vector would then have rotated twice around the origin.

Taking account of the denominator as well, it is seen that the two vectors $s - a$ and $s + \frac{\sigma'}{\gamma'}$, pivoting about points on the negative real

axis, will just cause a change of phase of (9.17), whereas the inverse system operator curve will encircle the origin no more than will its numerator vector locus just studied. Now in practice, the location of the roots of the characteristic equation is not known, but this inverse system transfer function may then be found from the results of a series of open loop frequency response tests by simply shifting the abscissa values in a plot of $\frac{\Psi_e}{\Psi_0}(j\,\omega')$, as suggested by equation (9.16). The condition that the curve of $\frac{\Psi_i}{\Psi_0}(j\,\omega')$ must not encircle the origin may therefore be replaced by the condition that the curve of $\frac{\Psi_e}{\Psi_0}(j\,\omega')$ must not encircle the -1 point of the real axis. It is also seen that the curve of $\frac{\Psi_e}{\Psi_0}(j\,\omega')$ will then pass to the left of this point when viewed in the direction of increasing ω'.

When the direct total open loop frequency response plot is preferred, the condition for a stable feed back system requires that this curve must not encircle the -1 point, and that it must leave this point to the left when viewed in the direction of increasing frequency. If the open loop curve has the character shown in Fig. 14 a, it is seen that it must pass inside the -1 point.

By use of the NYQUIST criterion, it is now possible to judge the amount of directional stability of the automatically steered ship, for which the open loop transfer functions at a few different combinations of control parameters were shown in Fig. 13 a. As has already been pointed out, the mathematical criterion for absolute stability will not be a sufficient criterion for a good control even in a technically calm sea, and the concepts of gain margin and phase margin have therefore been introduced in Fig. 13 b. The gain margin is often defined by the factor by which the open loop vector along the negative real axis, *i. e.* of 180° phase lag, may be increased without losing the absolute stability; it is then the reciprocal of the dynamic loop gain at the corresponding frequency. The phase margin is the phase shift from the negative real axis to that open loop vector, for which the dynamic loop gain is equal to unity. In general, a gain margin equal to $e^{1.2}$ and a phase margin of at least 30° are found suitable, this rule then making it sure that the magnification factor M does not exceed a value of 2—2.5 even at the resonant frequency. (For design purposes there

are some advantages of using a diagram, in which the logarithm of the magnitude of the open loop frequency response is plotted on a base of phase angle. If a point corresponding to the $(-1, 0)$ point of the vector diagram is taken to be the new origin, the gain and phase margins may at once be read from the intercepts on the axes.)

The full line curve in Fig. 13 a corresponds to the case with $\gamma' = \sigma' = 1$ and $t' = 0.1$, the stability of which has already been verified by the ROUTH criterion applied to the characteristic equation (9.18).

The three dotted curves all relate to a system, which has no rate control, but which is seen to be well stable for $t' = 0.1$, and just sufficiently stable for $t' = 0.5$. If the time lag is further increased to $t' = 1.0$ the system is certainly absolutely stable according to the NYQUIST criterion, but the stability margins are no longer acceptable. If a rate control is introduced, with $\sigma' = 0.5$, the stable performance will now be regained, however.

These examples will only give a rough idea of the methods and possibilities of controller design. For a fuller discussion of these questions, the reader is referred to the papers by SCHIFF and GIMPRICH (1949) and by RYDILL (1959). The latter paper in particular should be consulted when studying the additional complexities due to the operation in a seaway, in which case the functional relation between wave input and ship motion enters as a new block in the feed back diagram.

10. Stability and Manoeuvring Performance

In the deduction of the linear expression for the steady state radius of a gentle turn, equation (7.5), it was understood that the validity of the formula was limited to moderate curvatures and drift angles. According to the examples shown in Fig. 16, all from full-scale tests, the linear approximations do hold for manoeuvres with up to $20°$ rudder; if the steady state rudder turning moment may be estimated for still larger deflections, the non-linearities of the hull forces may probably still be assumed to be of secondary importance. The quotient $\left(\frac{R_c}{L}\right)_\delta \Big/ \left(\frac{R_c}{L}\right)_{20°}$ will then be characteristic for the type of rudder and stern arrangement used, and it is hoped that further information will be obtained from future routine trial trip tests. Additional full-scale data on rudder cavitation and flow break-down will also be welcome.

When comparing equation (7.5) with the criterion for dynamic

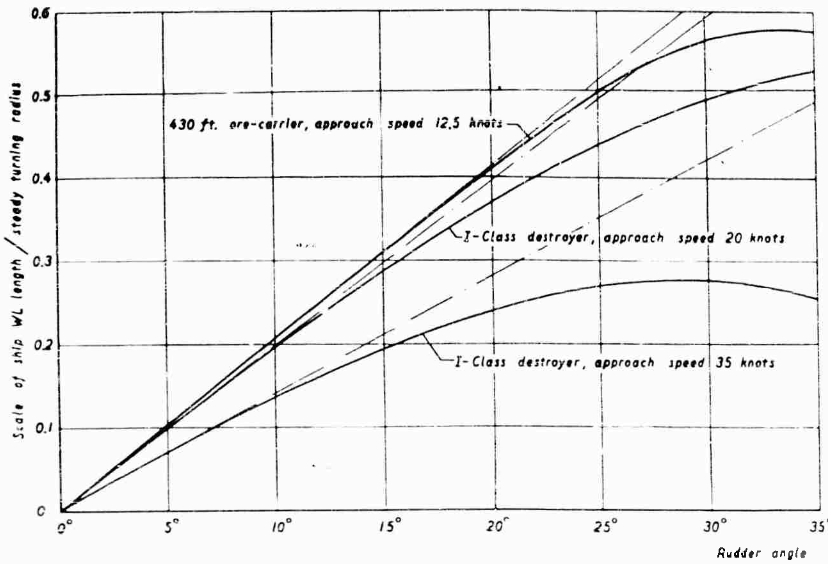


Fig. 16. Full-scale turning trial results showing validity and limitation of linear theory in formula (7.5). Destroyer data published by COLE 1938. Ore carrier data supplied by Kockums' shipyard in Malmö.

stability on a straight course, (8.3), it is easily seen that the stable ship will behave in a fully normal way, in as much as it will turn to starboard in response to the rudder being put to starboard. According to these same equations, a ship which is not stable on a straight course may tend to turn against the rudder. If when sailing with zero rudder, the latter ship is exposed to a small disturbance from its unstable equilibrium, it will enter a spiral to either side, the rate of turning steadily increasing until due to the non-linearities a stable equilibrium is reached in a turn of a certain radius R_{c_0}, *i. e.* at a turning rate equal to $r_{c_0} = \frac{180}{\pi} \cdot \frac{U}{R_{c_0}}$ °/s. (Fig. 17.) Again, if that ship is to be made to turn to starboard at a radius $R_c > R_{c_0}$, or at a rate $r_c < r_{c_0}$, this manoeuvre must be initiated by normal rudder but it is then necessary to check the ship with a reversed rudder. The new configuration of ship with port rudder is now dynamically stable in this gentle starboard turn. A plot of steady state rate of turning versus rudder angle, changed in steps from port to starboard (curve *c*) and back again (curve *d*), will then show up the hysteresis effects of Fig. 17, known as the DIEUDONNÉ spiral. In a sufficiently narrow turn the

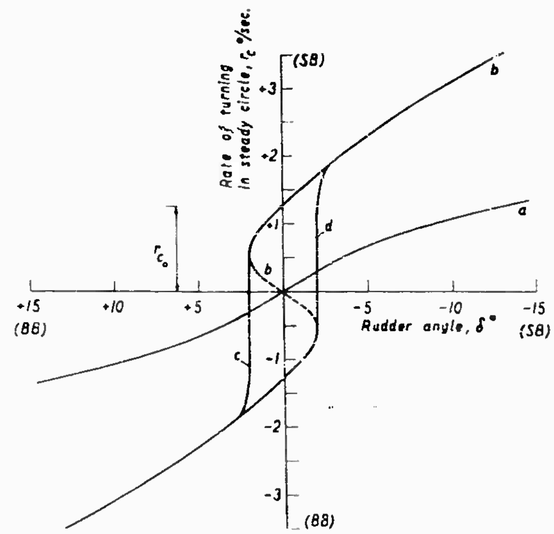


Fig. 17. The DIEUDONNÉ spiral. Hypotetical example showing the performance of a stable and an unstable ship.

ship will always behave like a stable one, and thus there are two regions within which the gradient $\partial r_c/\partial \delta_c$ is positive and finite. The vertical parts of the curves (c) and (d) correspond to an indifferent stability of the ship with slightly deflected rudder, moving in a curved flow. Steering along a straight mean course can be accomplished only by repeated use of the rudder.

The important role played by the non-linearities of the hull forces may be fully appreciated by studying the experimental curves of Figs. 4 and 6. It is easy to verify that the criterion of dynamic stability on a straight course is not satisfied for any of these model configurations, lacking deadwood area and rudder; on the other hand all of them will be stable in a curved motion, the radius of which is smaller for forms with a smaller length to beam ratio. By the addition of suitable stabilizing surfaces they may also be made dynamically stable in straight motion, in which case a relatively larger area is required for the ship with the smaller length to beam ratio. As an example, consider a series of destroyer forms, all with a beam to draught ratio equal to 3 but with varying length to beam. Let these forms be equipped with a fin or rudder of an effective aspect ratio equal to 2. To ensure a just dynamically stable ship, the common form with $L/B = 10$ will have a fin area of about 4 per cent of the

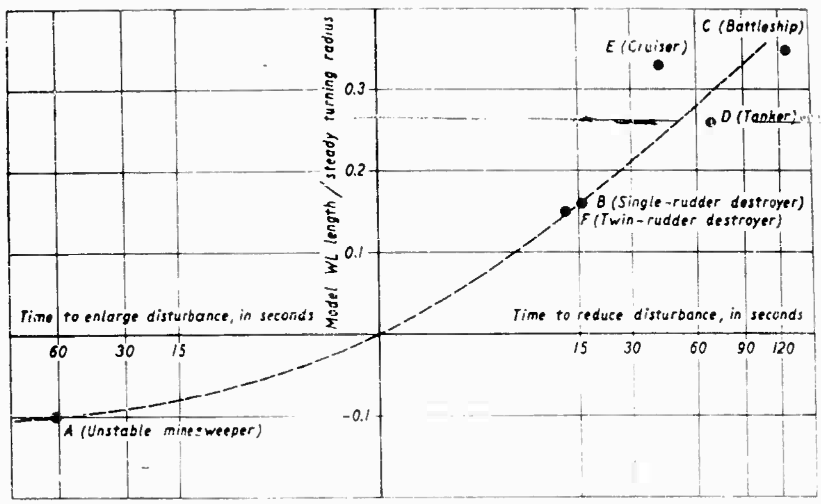


Fig. 18. Stability index and turning ability. The figure is based on model test data, published by DAVIDSON and SCHIFF 1946, and shows the non-dimensional turning circle curvature at a small turning moment coefficient, in all cases equal to -0.002, plotted versus the time, in which a small disturbance is reduced to e^{-1} of its initial value. ($F_{nL} = 0.24$.)

lateral area, whereas the shortened form with $L/B = 6$ will have at least 8 percent. In practice, less than half of the stablizing effect will be due to the rudder proper, however; most of it will be due to a long skeg, part of it also to the fin effect of the propellers.

A stern propeller in yawed flow is known to experience a pitching moment and a transverse force, which tends to stabilize the motion (GLAUERT 1919, RIBNER 1945). This effect is sometimes reduced by the straightening of the flow into the propeller due to the hull, sometimes it is partly compensated by a delay of the stabilizing boundary layer separation from a full stern due to propeller suction. Skegs, rudders and propellers add very little to the static (weathercock) stability of the hull. Their importance to course stability is to be found in their resistance to rotation, *i. e.* their damping effects present in the stability derivatives Y_r' and N_r'.

Once more turning to the equations (7.5) and (8.3), it is interesting to look for a relation between steady turning radius and the quantitative measure of dynamic stability, defined by the stability index p_1. In the paper by DAVIDSON and SCHIFF, the authors presented a diagram of the experimental product of minimum turning diameter and

rudder effectiveness on base of this index. (The rudder effectiveness was defined as relative rudder area times maximum deflection.) An unstable minesweeper model by chance fitted quite well in this diagram. A different presentation, which may be more suitable, is found in Fig. 18, where the rate of turning due to a constant small turning moment is shown on base of $-L/p_1 U$, *i. e.* the time, in which a small disturbance is reduced to e^{-1} of its initial value.

In Section 8 was shown how the exponential decay of a small initial disturbance in r or β could be calculated from the characteristic equation. If the ship is turning with a steady rate r_c and a drift angle β_c, corresponding to a rudder position δ_c, the transient phase of entering this turn may be calculated from the response to an initial disturbance, which is given by $-r_c$ and $-\beta_c$. (In the linear treatment, the stability derivatives must be valid for the hull with deflected rudder.) From this reasoning it is obvious that the more stable ship will need a shorter approach to a steady turning circle or change its course more willingly. This again serves to bridge the gap of conflicting demands for ease of steering and high manoeuvrability.

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12. Acknowledgement

The author is indebted to Dr. HANS EDSTRAND, Director of the Swedish State Shipbuilding Experimental Tank, for having been given the opportunity to undertake this study, as well as for the interest he has shown in it. Thanks are also due to the members of the staff of the tank who prepared the figures and typed the manuscript, and to Mr. P. D. FRASER-SMITH, who kindly revised the English text.

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